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DEVELOPMENT OF AIR-COOLED ENGINES WITH BLOWER COOLING

By Kurt Lohner

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DEVELOPMENT OF AIR-COOLED ENGINES WITH BLOWER COOLING*

By Kurt Löhner

This work contains the results of four hitherto unpublished researches of the D.V.L. (Deutsche Versuchsanstalt für Luftfahrt) (reference 1) during 1928-1932, in order to provide the basis for the development of air-cooled internal-combustion engines with blower cooling.

I. INTRODUCTION

Air-cooled aircraft engines have advantages over water-cooled engines, due to the elimination of the source of disturbance in the sensitive cooling plant. Moreover, the resistance of the ordinary radiators on water-cooled engines is great. The air-cooled engines in use are either radial or in-line and are cooled by the relative wind. This makes the cooling depend on the flight speed. Moreover, the air resistance is very great, especially at high speeds. This resistance can be reduced by suitable fairings (N.A.C.A. cowling, Townend ring), though these generally diminish the cooling effect. It is therefore difficult to increase the output of engines cooled by the relative wind.

In forced cooling, the conditions are more favorable. The air is then delivered, with the aid of blowers, at the places to be cooled. The velocity of the cooling current is largely independent of the flight speed and may even be greater than the latter. Since the air resistance can be smaller than in cooling by the relative wind, the power absorbed by the blower can be offset against the reduced air resistance. A favorable configuration of the engine can be obtained by the use of one or more blowers. Forced cooling is preferably used with in-line en-

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*"Die Entwicklung luftgekühlter Motoren mit Gebläsekühlung." D.V.L. Report 324. Automobiltechnische Zeitschrift, July 25 (pp. 351-357), and August 10 (pp. 375-383), 1933.

gines (fig. 1). If the current of cooling air enters the aircraft from the direction of flight and leaves it at the same velocity in the opposite direction, no air resistance ensues. In order to produce this result, the blower has only to offset the pressure loss from the deflections and frictional resistances in the engine, but does not have to furnish the power for generating the air velocity.

II. FUNDAMENTAL RESEARCHES ON THE POWER REQUIRED FOR COOLING FINNED CYLINDERS

1. Test Plant

The researches were based on triangular fins. Fins of uniform thickness, whose heat transmission is only slightly better, but which leave free a smaller cross section for the flow of the cooling air, were not used on account of their weight. As discovered by Schmidt (reference 2), fins of uniform heat-flow density require the least material. Here the temperature curve is a straight line from the temperature of the base to that of the air at the tip of the fin. The parts near the tip of the fin therefore show losses in energy without removing appreciable quantities of heat. Since the power expended for cooling must be kept small in an aircraft engine, the surfaces swept by the cooling air must be utilized to the best possible advantage. In order to obtain a short structural length of the engine, the enclosed space must be kept small. The fin of uniform heat-flow density does not therefore come into the question. The triangular fin is the most practical form, even as regards production.

A special heater was developed for use in determining the heat transfer from the shrouded and freely swept finned cylinders. The expenditure of power in forcing the air between the fins of the shrouded cylinder was also determined. The cylinders used for the investigation were made of pressed lautal. Their inside diameter was 135 mm (5.31 in.) and their height was 180 mm (7.09 in.). The diameter of the outside surface, to which the fins were attached, was 160 mm (6.3 in.). These dimensions were chosen because, for practical reasons, cylinders of 130 to 160 mm (5.12 to 6.3 in.) inside diameter are chiefly used in aircraft engines, a mean of 145 mm (5.71 in.), which corresponds to the outside diameter of 160 mm of the

test cylinders. Cylinders of smaller diameter have a smaller heat loading, so that they can be used without expensive cooling blowers. Cylinders of over 160 mm (6.3 in.) diameter are not made because their weight and bulk increase in an inadmissible manner.

Three test cylinders with different forms of fins were used. Form A had intervals of 8 mm (0.315 in.) between the tips of the fins; form B, 5 mm (0.197 in.); form C, 4 mm (0.157 in.) (fig. 2). The first tests were made with fins 42 mm (1.65 in.) high. The fins were then cut down to 28 mm (1.10 in.), then to 14 mm (0.55 in.), and finally to 7 mm (0.276 in.) in height. A further test was made with a smooth cylinder without fins.

The finned cylinder was mounted on a cast-silumin body with slide seat (fig. 3). This body had 16 holes of 14.3 mm (0.563 in.) diameter parallel to the axis, through which were stuck 8-mm (0.315-in.) silit rods with thickened metal-covered ends embedded at the bottom in an insulated contact plate and connected with the electric circuit. The upper and lower ends of the silumin body were insulated against the loss of heat by 20 to 30 mm (0.787 to 1.18 in.) of asbestos. The maximum heating power of the direct-current heating was 14 kW at about 100 volts and 140 amperes. The silit rods were heated to 1,200 to 1,400° C. (2,192 to 2,552° F.) and transmitted the heat to the inner surfaces of the holes in the silumin body, which in turn transmitted it to the test cylinder. The highest temperature allowable with regard to the strength characteristics of aluminum was 320° C. (608° F.).

The temperature of the finned cylinder was measured with 20 thermoelectric couples. Wires of copper and constantan of 0.3 mm (0.0118 in.) diameter, insulated with baked varnish and covered with silk, were electrically welded on the clean ends in hydrogen and inserted in radial holes of 1 mm (0.039 in.) diameter and 3 mm (0.118 in.) depth in the cylinder between two fins, thus affording good heat conduction from the cylinder to the head of the thermoelectric couple. The horizontal temperature distribution shows an increase in the flow direction of 30 to 80 percent of the temperature increase of the air. The temperature increase of the cylinder in the flow direction must be less than that of the air, because the heat conductivity of the cylinder militates against the occurrence of temperature differences. Moreover, the velocity of flow and the heat conductivity of the air increase with the temperature of the air.

The air for cooling the test cylinder is supplied by a centrifugal blower driven by a regulatable alternating-current motor. The cooling air is led through a pipe of 300 mm (11.81 in.) diameter, in which is installed a VDI Pitot tube of 120 mm (4.72 in.) diameter for measuring larger quantities of air, or of 80 mm (3.15 in.) diameter for smaller quantities.

The test cylinder was enclosed at the tips of the fins by a massive blackened shroud of hard wood (fig. 3). The top was formed by the metal cover of the heating body. The air was led from the air pipe to the shrouded cylinder through flat nozzles of a width corresponding to the height of the fins used. A short straight section was followed by a curved section to the height of the fins. The air flowed from the fins past the curved section into a collector having twice the width of the fins and five times their height. A metal plate, shaped like a comb with teeth projecting between the fins, extended to the end of the collector and insured a uniform flow of the air from both sides of the cylinder. The air velocity between the fins could be determined from the free cross section and the measured quantity of air. In the heat tests the reference temperature was the mean between the temperature of the air before contact with the heating cylinder and the temperature after the absorption of the heat transmitted to the cylinder.

After the removal of the shroud, the fins were subjected to a free jet of air. Due to the lower velocity and the smaller amount of heat transferred as the result of the nature of the centrifugal blower, it was necessary to work with lower heating powers than when the cylinder was shrouded.

The resistance of the nozzles between the air-delivery pipe and the shroud was determined by measurements in a free jet with relation to the quantity of air delivered and, in the evaluation of the tests, with relation to the total pressure. Amounts of $\frac{1}{2}$ to $3\frac{1}{2}$ percent of the total pressure had to be deducted.

The tests were made at various air velocities and heating powers. The mean temperature of the air and the temperature of the cylinder were therefore changed. In addition to variations in the velocity of the jet, variations in the density ρ, viscosity μ, and heat conductivity λ of the air were thus produced. The specific

heat c_p was nearly constant. Insofar as the heat dissipation sufficed, five different air velocities, each at three different heat stages, 4.5, 9, and 13.5 kW, corresponding to heat loadings of the cylinder jacket, 42,800, 85,600, and 128,400 kcal/m²h (15,779, 31,558, and 47,336 B.t.u./sq.ft./hr.), were investigated, as also the air resistance of the shrouded cylinder in the cold state.

2. Test Results

a) Representation of the test results.- The results of the experiments are shown in the figures. The air velocity w between the fins is

$$w = \frac{G\ R\ T_m}{p_b\ F_z}.$$

The velocity w_a of the heated air flowing from the collector to the cylinder is

$$w_a = \frac{G\ R\ T_a}{p_b\ F_a}$$

The heating power N is imparted to the air and heats it by the amount

$$T_a - T_e = 0.239\ \frac{N}{G\ c_p} = 0.988\ \frac{N}{G}$$

The difference between the temperature of the cylinder and that of the cooling air is

$$\Delta T = T_z - T_m = T_z - \frac{T_a + T_e}{2}$$

A cooling factor ϵ is used to designate the amount of heat transferred from the fins to the air. It represents the amount of heat per hour transmitted by the fins per square meter of the cylinder jacket of 160 mm (6.3 in.) diameter and at a temperature difference of one degree centigrade between the cylinder jacket and the air flowing past it.

$$\epsilon = \frac{860\ N}{F\ T\ h} = 9510\ \frac{N}{T}$$

The pressure drop Δp_o, which is necessary in the case of adiabatic flow for the production of the velocity v prevailing between the ribs of the cylinder, is

$$\Delta p_o = \frac{w^2 \rho}{2} = \frac{w^2 p_b}{2g R T_a}$$

The pressure reduction Δp_a for generating the outflow velocity w_a is

$$\Delta p_a = \frac{w_a^2 \rho}{2} = \frac{w_a^2 p_b}{2g R T_a}$$

The energy required for producing the velocity w between the cylinder fins in a constant flow is

$$L_1 = \Delta p_o V$$

In the heated body the volume V per second is based on the mean temperature between the cylinder fins. The total energy L actually required for producing the flow past the test cylinders is

$$L = (\Delta p_z + \Delta p_a) V_o.$$

Here Δp_z denotes the pressure reduction which takes place during the flow about the cylinder. In addition to the frictional and vortical losses about the cylinder, it also contains the vortical losses in the flow between the cylinder and collector in both directions. Δp_z is obtained directly from the experiments. The volume V_o per second is based on the cold air before contact with the cylinder.

In a similar process, employed for judging wind tunnels and the like, a "performance ratio" can be determined, in which the total energy L required for driving the air between the fins is expressed in relation to the energy L required to accelerate the air to the velocity at the cylinder. The energy ratio β represents the losses in the flow about the cylinder.

$$\beta = \frac{L}{L_1}$$

b) Heat transfer of test cylinder in a guided and in a free air stream.- The results of the measurements of the heat transfer of the enclosed test cylinder are shown in figure 4, which represents the cooling factor ϵ for the fins tested. ϵ is plotted against the velocity w of the cooling air between the fins of the cylinder. The

cooling factors ϵ, obtained with the three heat stages corresponding to 42,800, 85,600 and 128,400 kcal/m^2h heat loading of the cylinder, do not differ greatly. On the whole, the cooling factors are somewhat greater for low heating. It is worth noting that, even for high fins, the increase in the cooling factor ϵ, with increasing velocity of the cooling air, is still considerable. In figure 5 the cooling factor ϵ is plotted against the height of the fins for the cooling-air velocities of 30, 45 and 60 m/s (98.4, 147.6 and 196.8 ft./sec.) for the different forms of fins. The heat transfer increases with the height of the fins, but the increase soon becomes small, especially for slender fins of the form C at high velocities of the air. Two consecutive tests with a finless cylinder and a shroud 14 mm (0.551 in.) and also 7 mm (0.276 in.) from the cylinder surface, whereby in one case the cylinder was bright and in the other case was given a high radiation factor by an anodic "Specas" coating, showed that the effect of the radiation is negligible.

The differently shaped fins show considerable divergencies in heat transmission. This is chiefly attributable to two influences. The coefficient of heat transfer on the surface of the fins increases with the velocity of the air. This increases the temperature reduction from the base to the tip of the fins, which is especially great on long thin fins and has a detrimental effect on the heat dissipation. The second influence comes from the flow of the air between the fins. Fundamentally the heat-transfer coefficient, as based on the total area of the fins, is somewhat greater with narrow spaces between the fins than with wide spaces. Moreover, for sharply tapered fins with narrow intervals the flow velocity, due to air friction, is less at the base of the fins than at their tips. Hence the heat-transfer coefficient is smaller at the base of the fins. Furthermore the slowly flowing air is heated more at the base of the fins due to the longer period of contact, thereby unfavorably affecting the temperature difference.

In the experiments with the shrouded cylinder the quantity of air used for cooling is accurately known. Hence the actual mean air temperature can be determined. For the test cylinder exposed to the free flow the quantity of air involved in the cooling is unknown. In the calculation of the cooling factor the temperature of the air before the cylinder must therefore be introduced.

This temperature, however, due to the large quantity of air and the great turbulence, does not differ substantially from the temperature of the air near the cylinder. In figure 6 the cooling factor ϵ for the cylinder tested in the free jet is plotted against the velocity w of the cooling air. Since this velocity is small, the curvature of the lines is only slight. The temperature reduction from the base to the tip of the fins is not very noticeable. The improvement in the sweeping of the base of the fins with increasing air velocity, due to the elimination of the boundary layer, promotes the equalization. Of the three forms tested, the form C yields the greatest values for the coefficient of heat transfer. The comparison of forms A and B shows that, in a free air flow, the heat transfer is not increased by greatly reducing the spaces between the fins. The best value depends on the thickness of the fins and the velocity of the cooling air. In a free air flow the cooling factor ϵ depends on the heat loading. ϵ diminishes as the heat loading increases. The differences between the minimum and maximum heating stages run up to 10 percent of the cooling factor. In figure 7 the cooling factor for the cooling-air velocities of 12.5 and 25 m/s (41.0 and 82.0 ft./sec.) are plotted against the height of the fins, beginning with the smooth bright cylinder. The lines are only slightly curved. In comparison with the shrouded cylinder, the heat transfer of the exposed cylinder is only about two thirds.

c) Energy required for cooling the test cylinder.- The energy required for cooling the shrouded finned cylinder is determined from the pressure difference necessary to drive the air between the cooling fins and from the quantity of air used. With a given free cross section, this quantity is proportional to the velocity of the air. The total pressure in the heat tests serves, after deducting the nozzle losses, to accelerate the air and to overcome the air friction and the turbulence. It is difficult to separate these two quantities in the tests. From the quantity, temperature, pressure and cross section of the air, the velocity of the air leaving the collector and the pressure necessary to produce this velocity can be calculated. The pressure losses from turbulence produced by the fins and from the friction of the air on the cylinder can therefore be determined as the difference between the total measured pressure (after deducting nozzle losses) and the velocity head Δp_a at the outlet.

From these pressures, with consideration of the corresponding quantities of air, the energy ratio β can

be calculated. In figure 8, β is plotted against the cooling-air velocity w. β depends on the heat output, since both the pressure Δp_o, corresponding to the velocity at the cylinder, and the pressure Δp_a, corresponding to the outflow velocity, depend on the temperature of the air. In figure 8 the results of the experiments with the unheated cylinder are plotted. The scattering of the test points is here quite pronounced. Even the results of immediately consecutive tests show discrepancies. This indicates that the flow conditions were not quite stable.

From the curves of the test values for β at 100° C. (212° F.) air temperature and 1.035 absolute atmospheres in figure 9, is calculated the total pressure required to overcome the air friction and to accelerate the air by the amount indicated by the dynamic pressure in the outlet nozzle. These values are also valid for the system of air conduction employed in the experiments.

d) Relation between energy expenditure and heat transfer.- As shown by the heat-transfer curves, the heat transfer increases with increasing velocity of the cooling air. The energy required for forcing the air through the cylinders increases still more. Since, on an aircraft, this energy must always be derived from the engine, the least energy possible must be used for the cooling. Since, however, the engine output, thermal efficiency and reliability of operation diminish with increasing cylinder temperature, the best ratio between the cooling and the energy must be sought for every special case, with the utilization of the best form of fin.

For showing the relation between the degree of cooling and the energy required for cooling the test cylinder with air at 100° C. (212° F.) and 1.035 abs. atm., the heat transferred for a difference of 1° C. between the air and the cylinder shroud was determined from the experiments. It is expressed in kcal/h degrees. The energy required to force the cooling air between the cylinder fins is measured in mkg/s. It is necessary to distinguish between the total amount of energy expended in accelerating the air and overcoming friction and turbulence, and that required, at the given air velocity, simply for overcoming friction and turbulence.

In figure 10 the heat transfer Q_1 is plotted against the total energy expenditure for the fin forms tested. It was to be expected that the increase in the amount of heat

transferred would constantly diminish for a given form of fin with increasing energy expenditure. The longer fins are better than the shorter fins for cooling in the region investigated with uniform total energy expenditure. At high energy expenditure the thicker form B of the fins 42 mm (1.65 in.) long show the same rate of heat transfer as the thinner form C. This is due to the fact that the pressure required for forcing the cooling air between the fins of the form B is somewhat less than for the form C, and that, at high air velocities, the heat transfer from the thinner fins of the form C is no longer substantially better than from the thicker fins of the form B. Although the resistance of the flow of the cooling air and, despite the greater quantity of air used, the energy required for cooling with fins of the form A are considerably less than for the forms B and C, much more heat is transferred, for the same energy expenditure, with the forms B and C than with the form A, due to the great difference in the cooling factor ϵ. In figure 11 the amount of heat transferred is plotted against the height of the fins for 100, 200, and 400 mkg/s (723, 1,447, and 2,893 ft.-lb./sec.) total energy consumption. Even with a large expenditure of cooling energy and relatively high fins, we find an increase in the amount of heat transferred.

In figure 12 the transferred heat Q_1 is plotted against the energy required to overcome the resistance of the cylinder for the different forms of fins. The curves correspond to the curve for the total energy expenditure, but, aside from the smaller energy consumption, there are variations in the heat transfer between particular forms of fins.

2. Theoretical Investigations

a) Calculation of heat transfer by cylinders with fins.- If the coefficient of heat transfer α between the fins and the air and the temperature of the air over the whole surface of the fins were constant, the heat transferred by a cooling fin could be calculated. Such is not the case, however, of a cylinder exposed to a free air stream. This assumption is, nevertheless, approximately applicable to a shrouded cylinder for the most important forms of fins.

Schmidt (reference 2) calculated the heat transfer for a triangular fin on a flat surface with the aid of

Bessel functions. The calculation was made with the aid of a series development without the use of transcendental functions. Using the notation of figure 13, the heat transmitted by one half of the fin is

$$Q = - \alpha\, b\, T\, \Psi \left(\frac{\alpha\, b^2}{\lambda_a\, y_o} \right)$$

in which λ_a is the coefficient of heat conduction of aluminum. For half a fin of the diameter and height used in aircraft engines we obtain approximately

$$Q = - \alpha\, b\, T\, \pi\, D_m\, \Psi_1 \left(\frac{\alpha\, b^2}{\lambda_a\, y_o} \frac{D_m}{Di} \right)$$

The functions Ψ and Ψ_1 can be taken from figure 14 and enable a simple calculation.

The coefficient of heat transfer α between the surface of the fin and the air can be approximately determined from the equation found by Nusselt for the heat transfer in straight cylindrical tubes. According to Nusselt, allowance can be made for the cross-sectional shape by the introduction of the equivalent diameter $d_{ae} = \frac{4F}{U}$. If the heat transmitted by the cooling fins is thus determined, allowance must still be made for the cylinder surface between the fins.

On comparing the theoretical and experimental results, they are found, for the forms of fins most commonly used, to agree well enough for practical purposes. The calculated value is up to 15 percent too great, however, for very narrow spaces between the fins (forms B 28, B 42, C 14 and C 28).

b) Calculation of energy required to drive the cooling air between the fins.- For the shrouded cylinder, the space included between two fins, the cylinder and the shrouding may be regarded as a bent tube. In general the air enters at one end and leaves at the opposite end. If the curvatures at the inflow and outflow of the air are at first disregarded, we have two semicircular tubes of trapezoidal cross section between each two adjacent fins. The resistance to the flow in such a tube can be determined with close approximation.

According to Lorenz (reference 3) the pressure drop

for overcoming the turbulent wall friction in bent circular tubes is

$$\Delta p' = \frac{\varphi r_o}{d_{ae}} K w^2 \rho$$

in which φ is the angle of curvature of the tube, r_o the radius of curvature of the tube axis, $d_{ae} = 4F/U$ the equivalent tube diameter, ρ the air density and w the flow velocity. According to tests by Saph and Schoder, the factor k is

$$K = 0.1582 \left(\frac{\mu}{w\, d_{ae}\, \rho}\right)^{0.25}$$

According to Lorenz the pressure drop for the vortex formation is

$$\Delta p'' = \frac{\rho\, w^2\, d_{ae}}{2\, r_o} \frac{\varphi}{\pi}$$

Comparison of calculation and experiment shows that the curves have the same fundamental course. Calculation and experiment agree well for the fin form C. For the forms A and B, however, there are discrepancies in the absolute values. The calculated flow resistance is somewhat less than the measured, which is chiefly due to the simplifying assumptions regarding the inflow and outflow.

III. INVESTIGATION OF A LARGE SINGLE-CYLINDER ENGINE WITH RESPECT TO TEMPERATURE DISTRIBUTION, HEAT LOADING AND ENERGY EXPENDITURE FOR COOLING

1. Test Plant

The engine used for the investigation was a single-cylinder experimental engine of 160 mm (6.3 in.) bore and 220 mm (8.66 in.) stroke. Its output during the experiments was about 60 hp. at 1,500 to 1,900 r.p.m. It served for preliminary experiments for the development of a 9-cylinder radial engine of 550 hp. at full throttle. The cylinder head is roof-shaped and carries two intake valves and two salt-cooled exhaust valves. The valves and the

valve rockers are in a housing over the cylinder head and are actuated by push rods from the crankcase. The exhaust valves lie in the flow direction before the intake valves. The cast aluminum cylinder head is screwed, while hot, on to the open steel cylinder. At the cylinder head the fins are 12 mm (0.472 in.) apart and up to 30 mm (1.18 in.) high and 5 mm (0.2 in.) thick at the foot. On the cylinder body the fins are 8 mm (0.315 in.) apart, 9 to 22 mm (0.354 to 0.866 in.) high and 1.6 to 2.6 mm (0.063 to 0.102 in.) thick at the foot. In mounting such a radial engine on the airplane, the crankcase and cylinders can be shrouded to about the point where the aluminum head begins, the latter being exposed to the free air flow and the propeller slipstream.

The shrouding of the test cylinder was designed as it would be used on blower-cooled engines. Since the cylinder was designed for a radial engine with direct air cooling by the relative wind, of course not so good results can be obtained as with a cylinder especially designed for forced cooling. In particular, the distances between the fins and the free space between the control parts would have to be considerably smaller.

In the case of the cylinder exposed to a free air stream, the latter was directed against the cylinder by a centrifugal blower through a tube of 350 mm (13.78 in.) diameter (fig. 15). Since the outside diameter of the cylinder head was about 260 mm (10.24 in.) metal sheets were placed at the sides of the cylinder to prevent the air jet from being deflected laterally. The conditions were therefore similar to those of an exposed radial engine.

In the experiments with the enclosed cylinder (figs. 16 and 17) the air was driven, with the aid of the centrifugal blower, through an oval nozzle of about 100 mm (3.94 in.) width between the aluminum casing and the engine. A slot with rounded edges insured the smooth outflow of the air. Supplementary outflow openings were made in the intake chambers.

The first shroud was everywhere 10 mm from the edges of the fins. The second shroud touched the edges of the fins. The third shroud was like the second, excepting that a wooden block was mounted between the intake valves, which strengthened the air flow between the exhaust and intake valves. This block was screwed to the cam case

from below, touched the fins of the cylinder head and was 10 mm (0.394 in.) from the intake chambers. The crankcase was cooled by a fan. The lubricating oil was forced through a copper coil and cooled with water.

The output of the single-cylinder engine was determined by a water brake. The temperature of the cylinder was measured at 33 places with thermoelectric couples of insulated copper and constantan wires of 0.34 mm (0.0134 in.) diameter. The heads of these couples were set 3 mm (0.118 in.) deep in aluminum and peened, on the steel jacket, with a piece of sheet metal which was insulated by asbestos against loss of heat. There were 8 thermoelectric couples on the cylinder jacket and 25 on the head. The test points were quite uniformly distributed over the cylinder head, including all the important parts (spark plugs, valve seats and upper part of cylinder). After a number of preliminary tests, the amount of air delivered by the blower and the air velocity at the cylinder were measured with a Bruhn nozzle. For the shrouded cylinder, the total pressure at the end of the air tube served to determine the resistance to the air flowing past the cylinder. The resistance of the oval intake nozzle was determined separately and subtracted from the total resistance.

2. Temperature Distribution

At the beginning of the investigation a preliminary test was made with the single-cylinder engine in a free air jet. As shown by figure 18, the operating temperature was reached in four to six minutes after full throttle.

The results of the tests on the temperature distribution are shown in figures 19 to 21. The figures give the temperature differences between the indicated points and the air before the cylinder from the middle axis out. The engine produced 60 hp. at 1,700 r.p.m. and was exposed to an air flow of 45 m/s (147.6 ft./sec.). Figure 19 shows a vertical section through the center of the cylinder in the direction of flow. The temperatures of the cylinder liner are lower in front than behind. The fact that, on the contrary, the temperatures on the cylinder head are higher in front is chiefly due to the location of the exhaust valves.

Figure 20 shows a section through the axis of the cylinder perpendicular to the direction of flow. The temperatures on the right and left sides do not differ materially. Of course the temperature of the outlet chambers is considerably higher than that of the inlet chambers.

Figure 21 shows a horizontal section through the cylinder head at the height of the spark-plug holes. The maximum temperatures are forward laterally and at the spark plugs. This is due to the location of the exhaust which causes additional heating of the neighboring parts of the cylinder head while, on the rear side of the cylinder, the intake has a certain cooling effect. The shrouded cylinder, in which the air is forced past the cooling fins, has lower temperatures throughout than the cylinder exposed to the free air jet.

3. Energy Required to Cool Single-Cylinder Engines

For every shrouding and every air speed, three tests were made at about 1,500, 1,700 and 1,900 r.p.m. The tests at 1,500 and 1,900 r.p.m. simply served for control of the test at 1,700 r.p.m. which alone was evaluated. The mean value of the indications of the 25 thermoelectric couples was taken as the temperature of the cylinder head. The temperature of the hottest places, which lie on both sides of the cylinder head between the outlet and inlet chambers, and the mean temperatures of the steel cylinder are correspondingly represented. The air velocity was reduced from the maximum velocity produced by the centrifugal blower as far as was consistent with reliability of operation.

The resistance of the exposed cylinder head is known from the researches of the Aerodynamic Institute of the Aachen Polytechnic School. The energy required for cooling can therefore be determined along with the air velocity. For the shrouded cylinder, it is obtained from the measured quantity of air blown through the shroud and from the pressure required (fig. 22). Thereby two different energy requirements must be distinguished according to how the engine is used (fig. 23). When the engine is stationary, the energy requirement for accelerating the cooling air and for overcoming the resistance between the cylinder and shrouding must be greater. On an airplane in flight, the energy for accelerating the cooling air can be entirely or partially dispensed with, if the air is taken from the surrounding space at the existing relative velocity and released at the same velocity in the direction of flight. Theoretically the same result could be obtained through the use of diffuser nozzles with the engine stationary, but the efficiency of such nozzles is too poor.

For effecting the energy balance of the engine, the efficiency of the cooling-air blower must be taken into consideration. In the use of centrifugal or axial blowers it can be taken at 50 to 70 percent. Similarly for a radial engine on an airplane the propeller efficiency can be taken at 60 to 80 percent, from which must be deducted the structural drag of the fuselage, etc., due to the projecting cylinder heads, so that on the whole the efficiency is only about the same as for the production of the cooling-air current.

The results of the test runs (figs. 24 and 25) show that the temperature of the aircraft engine and the energy required for cooling can be reduced by suitable shrouding and cooling-air control by blower. Moreover, it is possible to conduct the air to the points of maximum heat loading, so as to avoid excessive temperatures at particular points. This gives the designer a free hand in determining the form of the engine.

The data used for calculating the energy required for cooling a radial engine exposed to a free air flow are based on tests made with a cylinder by the Aerodynamical Institute of the Aachen Polytechnic School. It was found possible, by a better design, especially of the exhaust stacks, and a few cooling fins, to reduce the air resistance about 20 percent. It may also be expected that the temperature of the cylinder head will be somewhat further reduced by improving the conduction of the air current. On the other hand, the resistance of the radial engine will be somewhat increased by using outlet tubes, since the drag measurements without exhaust pipes were made only with short exhaust stacks.

The temperatures of the cylinder head and lining could be considerably reduced on an engine developed for blower cooling. Thinner fins and narrower intervals would be of great advantage which, with the same free cross section, would increase the heat-emitting area and the heat-transfer coefficient. The free cross section between the right and left valve chambers might be made smaller and the cooling of the space between the intake and exhaust valves improved. With the same energy expenditure for cooling, the engine might thus yield considerably higher performances.

The measurements also enable the determination of the cooling for the shrouded radial engine cooled by the rela-

tive wind, if the positive pressure in front of the inflow opening and the negative pressure behind the engine at the outflow opening for the cooling air are measured. This can be done with a model of the shroud in the wind tunnel whereby the flow resistance is regulated by screens.

4. Heat Loading and Heat Transfer of Single-Cylinder Engine

While an aircraft engine is running, heat is imparted to the walls by the working gases. The amount of heat transferred depends on the temperature of the gases and walls, as well as on the coefficient of heat transfer, the area and the length of time the gases remain in contact with the walls. The coefficient of heat transfer is very large during combustion, but smaller during expansion and exhaust. During the intake stroke the mixture absorbs heat from the walls of the compression chamber and from the piston, since these are continuously exposed to the working gases, while the cylinder liner is protected part of the time by the piston from contact with the working gases. The heat imparted to the valves must be transmitted to the valve seats and pipes. The exhaust valve is strongly heated by the swiftly flowing exhaust gases. The determination of the heat imparted to the valves has hitherto been impossible, because the conditions of heat intake and outgo have been so varied. The exhaust stacks are heated by the outflowing gases, while the intake valve, on the contrary, is cooled by the cold inflowing mixture. The heat absorbed by the piston head is almost as much as that absorbed by the combustion chamber, the temperatures being also of the same order of magnitude. The heat absorbed by the piston and that produced by the friction of the piston is partially imparted by the piston rings to the cylinder walls and partially to the oil and air in the crankcase. Since the cylinder liner and piston rings are usually somewhat cooler than the upper part of the cylinder, there is a slight heat transfer from the upper to the lower part of the cylinder.

The walls of the "Neonalium" cylinder head are very thick and transmit considerable heat which tends to equalize the temperature. The temperature of certain very hot spots (e.g., between the exhaust valves and between the intake and exhaust valves) is thus reduced, as well as the effect of the heat transfer by special cooling fins. The threading between the cylinder head and the steel cyl-

inder extends down over the upper part of the cylinder. An axially directed flow of heat is produced in this light-metal cover. The cooling of the lower part of the cylinder is somewhat increased by the lubricating oil.

The heat transfer by the fins of a shrouded cylinder can be determined from the results of the investigation of the cooling of finned cylinders, if the temperature of the cylinder is known. The experiments described in section III,2, therefore enable the determination of the amount of heat imparted to the air by the shrouded single-cylinder engine through the determination of the heat transferred by each fin. In the calculation of the heat transferred by the cylinder head, allowance must be made for the fact that the exhaust stacks absorb a considerable quantity of heat, only a small part of which comes from the combustion chamber, the larger part coming from the exhaust gases. In the intake pipes, on the other hand, heat is absorbed by the cold inflowing gases and carried into the combustion chamber. The amount of heat absorbed by the cylinder walls steadily decreases toward the lower piston dead center.

The following quantities of heat are transferred:

1. Cylinder head to upper edge of piston at top dead center, i.e., the beginning of the steel liner ..... 5,618 kcal/h (22,293 B.t.u./hr.)

2. Two exhaust stacks, from point of contact with cylinder head ..... 3,200 kcal/h (12,698 B.t.u./hr.)

3. Two inlet pipes, from point of contact with cylinder head ..... 760 kcal/h (3,016 B.t.u./hr.)

4. Cylinder, upper part with lower portion of cylinder head head ..... 6,277 kcal/h (24,909 B.t.u./hr.)

5. Cylinder, lower part with steel fins ..... 1,665 kcal/h (6,607 B.t.u./hr.)

The total amount of heat given off by the cylinder is therefore 17,520 kcal/h (69,524 B.t.u./hr.). The engine output was 60 hp. There were therefore 292 kcal/hp/h (1,159 B.t.u./hp./hr.) imparted to the cooling air. This value seems probable on comparison with the results of experiments with high-class, water-cooled aircraft engines, in which, due to lower wall temperatures, somewhat greater quantities of heat were carried off. It is still to be taken into account that the housing of a single-cylinder engine carries off somewhat more heat than that of a radial engine. It is worth noting that the two exhaust stacks transmit 18 percent of the total amount of heat.

The heat flow from the cylinder head to the liner can be calculated from the cross section of the aluminum ring and the temperature drop known from the tests. The cross section is 108 cm^2 (16.74 sq.in.), the temperature drop 39° C. (70.2° F.) for 48 mm (1.89 in.) length, and the amount of heat carried off, for a heat conductivity of aluminum of 125 kcal/m^2/h (46.08 B.t.u./sq.ft./hr.) is therefore 1,090 kcal/h (4,325 B.t.u./hr.). All together therefore, the cylinder head transmits 10,668 kcal/h (42,333 B.t.u./hr.); the upper part of the cylinder liner, 5,187 kcal/h (20,583 B.t.u./hr.); and the lower part of the liner, 1,665 kcal/h (6,607 B.t.u./hr.).

The mean value of the heat loading of the inner wall of the compression chamber can be determined. The inner surface of the compression chamber, including valves but not piston, is 393 cm^2 (60.9 sq.in.); of the two intake valves, 48 cm^2 (7.44 sq.in.); of the two exhaust valves, 40 cm^2 (6.2 sq.in.). Disregarding the axial heat flow in the aluminum part of the cylinder, the heat loading of the compression chamber is therefore 244,000 kcal/m^2/h (89,954 B.t.u./sq.ft./hr.); or, including the axial heat flow, 272,000 kcal/m^2/h (100,276 B.t.u./sq.ft./hr.).

For the part of the cylinder on which is screwed the aluminum head, we obtain, on the basis of the heat transfer by the cooling fins, 199,000 kcal/m^2/h (73,363 B.t.u./sq.ft./hr.) at the top and 145,000 kcal/m^2/h (53,456 B.t.u./sq.ft./hr.) at the bottom. The amounts of heat transmitted by the individual fins are:

1st fin,	878	kcal/h	= 145,500	kcal/m^2/h
2d "	945	"	= 156,500	"
3d "	1,015	"	= 168,000	"
4th "	1,085	"	= 180,000	"

5th fin,	1,154 kcal/h	= 191,500	kcal/m^2/h
6th "	1,200 "	= 199,000	"
7th "	1,250 "	= 207,000	"

(kcal/h × 3.96825 = B.t.u./hr.)
(kcal/m^2/h × 0.368663 = B.t.u./sq.ft./hr.)

It must be taken into consideration that there is an axial flow of heat inside the aluminum jacket of the upper part of the cylinder. The actual heat loading of the inner surface of the cylinder must therefore be such that the lowest portion of the cylinder head will have the same heat loading as the steel cylinder and that the upper edge of the steel cylinder will have the same heat loading as the cylinder head.

The heat loading of the liner at the top dead center of the piston would likewise have to be just as great as that of the cylinder head with uniform heat transfer. If, in the heat transfer of the cylinder head, the exhaust and intake pipes are disregarded and the valve surfaces are therefore left out, then 5,618 kcal/h (22,294 B.t.u./hr.) will be transferred for 305 cm^2 (47.27 sq.in.) of surface. There is therefore a heat loading of 184,500 kcal/m^2/h (68,018 B.t.u./sq.ft./hr.). To this must be added 1,090 kcal/h (4,325 B.t.u./hr.) removed from the cylinder head by the axial heat flow, which, for 393 cm^2 (60.91 sq.in.) of the head (including valves), corresponds to a heat loading of 27,800 kcal/m^2/h (10,249 B.t.u./sq.ft./hr.). This makes the total heat loading of the cylinder head 212,300 kcal/m^2/h (78,267 B.t.u./sq.ft./hr.), which agrees very well with the value obtained at the upper edge of the liner.

The quantities of heat transferred by the steel fins, including the adjacent cylinder surfaces, are:

1st steel fin (bottom)	101 kcal/h	= 25,300	kcal/m^2/h
5th steel fin	104 "	= 25,900	"
10th " "	114 "	= 28,600	"
14th " " (top)	124 "	= 30,300	"

The heat loadings in figure 26 were plotted on the basis of the experiments. The heat loading is plotted vertically over the inner wall of the cylinder. The quantities of heat are not perpendicular to the bent lines. Curve I represents the mean heat loading of the cylinder

head disregarding the axial heat flow with inclusion of the heat transfer by the intake and exhaust pipes. Curve II takes into account the axial heat flow in the cylinder-head threading and likewise contains the heat given off by the intake and exhaust pipes. Curve III contains the heat transferred by the cylinder head without the heat obtained from the intake and exhaust pipes and without the axial flow, as well as without taking the valve surfaces into consideration. Curve IV contains the heat transferred by the cylinder head including the axial-heat flow in the cylinder-head threading, but without taking into account the heat from the intake and exhaust pipes and the valve surfaces. Curve V, joining curve IV and representing the heat loading of the cylinder-head threading, was determined from the known beginning and ending points and the area. Curve VI represents the quantity of heat transferred by the fins of the cylinder-head threading. Curve VII represents the heat loading of the steel cylinder.

The quantity of heat transferred to the valve connections and carried off by them cannot be added directly to the heat loading calculated for the inner surface of the cylinder head, because the paths for the heat flow do not lie immediately in the walls of the compression chamber. Curves IV, V, and VI therefore correspond to the quantities of heat transferred by the working gases at every point in the cylinder, i.e., to the actual heat loading of the cylinder. Reliability of operation naturally also depends largely on quantity of heat transferred to the exhaust stacks.

The heat balance of the engine can be determined from the experiments. At the compression ratio of 5.8 the mean fuel consumption was 235 g/hp/h (0.52 lb./hp./hr.) for a mixture of 80 percent benzol and 20 percent gasoline, the heat value of which can be put at 9,700 kcal (38,492 B.t.u.). The heat intake was therefore 2,280 kcal/hp/h (9,048 B.t.u./hp./hr.). The mean effective pressure was 7.2 kg/cm^2 (102.41 lb./sq.in.) at 60 hp. and 1,700 r.p.m., corresponding to a piston speed of 12.5 m/s (41.0 ft./sec.). The mechanical friction can be represented by a mean frictional pressure of 0.8 kg/cm^2 (11.38 lb./sq.in.) (reference 4), and the flow resistance for intake and exhaust by 0.6 kg/cm^2 (8.53 lb./sq.in.). The frictional heat is partially included in the heat transferred to the cylinder, which is not considered in the balance. The individual items of the heat balance are generally expressed with relation to the quantity of heat taken in with the fuel. With a car-

buretor engine it is more practical, however, to compare them with the effective power, since an increase in the specific fuel consumption above a certain amount (about 5 percent excess fuel) causes no noticeable change in the amount of heat transferred, but only increases the proportion of unburned gases in the exhaust. In the first numerical column of the following table are given the number of calories per horsepower-hour; in the second column, their relation to the effective power; and in the third column, to the heat introduced with the fuel.

	kcal/hp/h	percent	percent
Effective power	632	100.0	27.7
Mechanical friction	70	11.1	3.0
Flow resistance	53	8.3	2.3
Heat transfer in cylinder	292	46.0	12.8
Heat transfer in oil, about	29	4.6	1.2
Exhaust gas and unburned	1,204	191.0	53.0
Total heat in fuel	2,280	361.0	100.0

(kcal/hp/h × 3.96825 = B.t.u./hp./hr.)

IV. EXPERIMENTS ON A TWELVE-CYLINDER V-ENGINE WITH COOLING BLOWERS

1. Design and Purpose of Engine

The air-cooled 12-cylinder engine was built in 1928 to 1930 for test purposes. Six centrifugal blowers were installed between the banks of the 12-cylinder V-engine (figs. 27-29) in such fashion that two opposite cylinders are cooled by each blower. The cooling air flows from above to the two lateral intake openings of each blower and is conducted to the cylinders by guide vanes. The blower rotors are mounted on a common shaft with two bearings. The coupling with the driving gear forms a third bearing.

The separate cylinders of forged chrome-nickel steel have fins of uniform height from the top to the cylinder flange and are sunk very deep into the crankcase. The top of the closed bell-shaped cylinder is very thick and is secured to the aluminum head by 14 bronze screws. The cylinder heads are separate and carry a common camshaft. They each have four valves parallel to the cylinder axis. The intake connections are inside the V, while the exhaust connections are on the outside. Ignition is effected by two lateral spark plugs. Temporary increase of the starting power and the maintenance of the power at a moderate altitude is provided for by a removable centrifugal blower which is designed to increase the suction pressure by 20 percent.

The dimensions of the engine are: bore, 155 mm (6.10 in.); stroke, 210 mm (8.27 in.); piston displacement, 47.6 liters (2,904.7 cu.in.); compression ratio, $\epsilon = 5.5$; 4 Zenith carburetors 75 IR; gear ratio between crankshaft and blower shaft, 1:2.19.

The full power of the engine is 700 hp. at 1,800 r.p.m. A continuous output of 560 hp. at 1,670 r.p.m. can be maintained with good reliability of operation. With the blower running, a starting power of 840 hp. at 1,910 r.p.m. can be attained for a brief period. The required powers were mostly attained; the power with the blower, exceeded. The cooling, however, presented difficulties.

2. Development of Suitable System of Air Conduction

a) Original system.- The engine power was determined by means of a 4-blade calibrated brake propeller of 2.13 m (6.99 ft.) diameter. The cylinders were not enclosed. The outer side of the cylinders was cooled by the slipstream. After extensive preliminary tests by the manufacturer, two acceptance runs were conducted by the D.V.L. in the factory, in which tests the most important temperature measurements were made for the first time. The most important results of these tests are given in the table.

CYLINDER TEMPERATURES OF THE AIR-COOLED 700 HP. ENGINE

WITH COOLING BLOWERS WITH THE ORIGINAL AND WITH THE DVL SHROUDING

(Temperature Differences between Air and Engine)

Test point	Between 3d and 4th fin from bottom				Between 3d and 4th fin from top				Between 17th and 18th fin from top				Exhaust stacks												Spark plugs								Starter connections						
	front	inside	rear	outside	front	inside	rear	outside	front	inside	rear	outside	outside	Between the stacks							under, front	under, rear	over, rear		rear	rear	front	front	rear	front	rear	front	rear							
Cylinder No.	3	8	8	2	8	8	8	2	8	8	8	2	2	4	6	7	8	9	10	11	12	8	8	8	2	4	5	8	8	9	10	11	12	1	2	4	8	10	12
Original cowl																																							
Brake propeller																																							
Ne = 560 hp.	110	139	103	304	277	203	123	215	314	313	92	83	115	171	270	215	235	235	250	287	330	316	290	245	295	194	220	237	280	240	230	227	245	265	225	180	277	218	284	289	290
nM = 1690 r.p.m.																																							
nG = 3720 "																																							

									1
									DVL cowling, pusher propeller, N_e=560 hp., nM = 1690 r.p.m., nG = 3720 "
Ring under start of threaded connection							Cylinder No.		
							1		281
							2		317
							3		287
							4		305
							5		-
							6		281
							7		200
							8		-
							9		225
							10		238
							11		321
							12		314
Ring under spark plug							11	front	-
							11	rear	210
							5	front	-
							5	rear	225
Cylinder head	outlet						11	between	258
							5	between	-
							11	rear, over	205
							5	rear, over	246
	inlet						11	between	155
							5	between	145
Cylinder cover outside between							11	11th & 12th fin from top	351
							5		370
							11	4th & 5th fin from bottom	180
							5		184

2	3	4	5	6	
DVL cowling, pusher propeller, N_e=510 hp., nM = 1575 r.p.m., nG = 3460 "	DVL cowling, pusher propeller, N_e=510 hp., nM = 1915 r.p.m., nG = 4200 "	DVL cowling, tractor propeller, N_e=350 hp., nM = 1416 r.p.m. nG = 3200 "	DVL cowling, pusher propeller, N_e=560 hp., nM = 1750 r.p.m. nG = 4900 "	DVL cowling, pusher propeller, N_e=620 hp., nM = 1808 r.p.m. nG = 5063 "	
-	-	-	234	281	
300	285	300	275	317	
290	270	305	254	287	
270	270	315	258	305	
310	295	330	298	-	
235	260	-	269	281	
260	240	270	179	200	
235	225	255	230	-	
-	-	-	220	225	
220	220	310	280	238	
220	220	305	274	321	
-	-	-	258	314	
-	-	-	191	-	
165	160	160	179	210	
195	185	205	-	-	
-	-	-	183	225	
260	270	285	235	258	
310	320	305	283	-	
200	210	205	174	205	
215	250	220	198	246	
105	105	125	108	155	
130	130	150	125	145	
250	260	305	315	351	
310	295	315	304	370	
165	175	165	182	180	
-	-	-	159	184	

After removal of the engine, further test runs were made by the D.V.L. Temperature measurements showed that the cooling of the engine was insufficient. Another indication of this was found on dismantling the engine. Several valve heads and the heads and upper plates of the cylinders were warped, the curvature amounting to as much as 0.4 mm (0.016 in.). The temperatures of the cylinders and cylinder heads must accordingly have been above the permissible limit even at 560 hp. The engine was put in shape for further use only by scraping the cylinder heads and replacing the damaged valves.

b) Free blower with guides.- The most important task was to guide the cooling air so as to cool adequately the most strongly heated parts of the engine. The cylinder liner must be kept as cool as possible, in order to maintain low piston temperatures and good functioning. For this purpose tests were made in which the driving gear between the crankshaft and blower shaft was removed and the latter was connected directly with a cradle dynamometer.

In order to determine the properties of the free blower (fig. 29), a test run was first made without the engine cylinders. The power absorbed by the blower was 6.8 hp. at 2,500 r.p.m., 12.6 hp. at 3,000 r.p.m., and 17.4 hp. at 3,460 r.p.m.

c) Cylinders without guides.- Engine, cylinders and blower were again assembled and the blower shaft was driven by the cradle dynamometer. The velocity of the air leaving the spaces between the cylinders and between the fins of the cylinder heads was measured at 3,600 r.p.m. The results of the determination of the velocity distribution are plotted in figure 30.

The velocity of flow is very uneven both on the sides of the engine and over the individual cylinders. The flow between the cylinders is essentially rectilinear. The cooling of the blower side and of the cylinder parts opposite one another is therefore good, but the outer side of the cylinders is not touched by the cooling air from the blower and is consequently overheated by the engine. This can be only partly remedied by guiding the relative wind. The power absorbed in driving the blower amounts to 15.2 hp. at 3,600 r.p.m. In order to determine the quantity of air delivered by the blower, experiments were performed, in which an air-tight hood with a VDI measuring nozzle of 120 mm (4.72 in.) diameter was placed over the intake ori-

fice of the cylinders between which the measurements were made. The adjacent intake openings of the blower were provided with similar hoods, in order to effect a corresponding throttling. The nozzle and hoods produced a negative pressure before the blower, which reduced the quantity of air to two thirds of its value on a conventional engine.

The amount of cooling air drawn between cylinders 2 and 3 at 3,600 r.p.m. was 0.76 kg/s (1.68 lb./sec.); between cylinders 3 and 4 it was 0.91 kg/s (2.01 lb./sec.). The total amount of air used for cooling an uncowled engine was accordingly calculated to amount to 4.7 kg/s (10.36 lb./sec.).

d) Air conduction about the cylinders.- The following experiments served to develop a system of air conduction in which, with the best possible utilization of the cooling air delivered by the blower, all parts of the engine were cooled according to their heat absorption. The air was therefore conducted, on the basis of previous investigations, by guiding surfaces and cowlings between the cylinder fins and over the cylinder heads.

Since the test runs showed that the cylinder heads were damaged by overheating due to the reduced ventilation with the original cowling, no cowlings were used. Moreover, the cooling was improved by conducting an air current upward from the base of the cylinder through the triangular passage formed by the blower wall and the cylinder cowlings with the use of air which was saved by the fairing at the cylinder foot. The cylinder cooling was improved by conducting the cooling air around it.

The quantity of air flowing between the individual cylinders was found by measuring the inflow and outflow cross sections of the air conduits. The cross sections were made so large at the top of the cylinders that the flow of the cooling air was not impeded, but they were conically tapered toward the bottom of the cylinders, in order to reduce the cooling air to 1/2 or 1/3 of the amount with uncovered cylinders. Air-flow measurements showed the need of better cooling for the spark plugs and for the fins lying immediately behind them. This was accomplished by folding back the guide sheets around the spark plugs. Figure 28 shows the inner side of the cylinders with blowers removed, while figure 27 shows the outer side.

The results of the tests are shown in figure 31. The velocity of the cooling air along the cylinder was somewhat greater than with the previous shrouds. The cooling of the cylinder lining is adequate, due to the greater cooling effect of the flow around the whole cylinder. The air velocities on the two sides of the cylinders are still unequal, even with this shrouding, though the discrepancies are not excessive. No improvement can be effected without reconstruction of the blowers.

The air measurements at 3,600 r.p.m. yielded 0.7 kg/s (1.54 lb./sec.) between cylinders 2 and 3 and 0.82 kg/s (1.81 lb./sec.) between cylinders 3 and 4, corresponding to a total of 4.32 kg/s (9.52 lb./sec.) for the engine. This shows a saving of about 10 percent of the air as compared with the exposed cylinders. The power developed at 3,600 r.p.m. was 13.8 hp.

3. Temperature Measurements on Running Engine with Two Different Systems of Air Conduction

In the acceptance runs at the factory, temperature measurements were made on the engine fitted with the original air guides. The engine was equipped with the brake propeller and developed in the tests 560 hp. at 1,690 r.p.m. The table shows the difference between the temperature of the air on entering the blower and the temperature of the test point.

The measurements show very great temperature differences between the inner and outer sides of the cylinders, the latter being too high for reliable operation of the engine. The cylinder-head temperatures near the exhaust approach the upper limit, even exceeding it on certain cylinder heads, so that, at full load and with the use of superchargers, there is danger of warping the cylinder heads, cylinder plates and valves.

Following the development of the DVL system of air conduction the engine was mounted on the torque stand and fitted with thermoelectric couples. In order to eliminate the effect of the propeller slipstream, the brake used was a 4-blade DVL adjustable-pitch pusher propeller. The test results are likewise shown in the table and in figure 30. The temperatures at 560 hp. and 1,690 r.p.m., reached the upper limit and in some cases exceeded it, so that there was danger of disturbances in protracted operation. The

mean temperature of the engine is hardly any lower than with the original system of air conduction, but the temperature distribution is considerably more uniform. The spark-plug temperature of 190° C. (374° F.) may be regarded as low, the temperature of 240° C. (464° F.) with the original air conduction being rather high.

Two supplementary tests (see table) with steeper and flatter pitch of the brake propeller give an idea of the possibilities of improvement by increasing the revolution speed of the engine and blower with the same engine power. The engine ran with 510 hp. effective output at 1,575 and 1,950 r.p.m. The increase in the r.p.m. necessitates, despite constant effective output, an increase in the amount of heat transferred from the combustion gases to the walls. The simultaneous increase in the blower r.p.m., however, improves the cooling sufficiently to reduce the mean temperature of the engine. A brief comparative test with the 4-blade brake propeller showed that the cooling effect was very greatly impaired by the unfavorable air flow.

4. Increase in Reliability of Operation from Increased Cooling

In the blower tests with the final form of the DVL system of air conduction, 4.32 kg/s (9.52 lb./sec.) of air was delivered at 3,600 r.p.m. of the blower. The normal engine speed for a constant output of 560 hp. was 1,695, corresponding to 3,710 r.p.m. of the blower. The amount of air delivered was calculated as 4.45 kg/s (9.81 lb./sec.) and the energy absorbed as 15.1 hp., which was 2.7 percent of the engine power.

In a normal aircooled aircraft engine, about 320 kcal/hp/h (1,270 B.t.u./hp./hr.) must be removed for medium loading. In a 560 hp. 12-cylinder engine, therefore, about 170,000 kcal/h (674,602 B.t.u./hr.) would need to be removed. The temperature of the cooling air would therefore be raised about 46° C. (114.8° F.)

In order to increase the reliability of operation of the engine with retention of the form under investigation, there remains the possibility of increasing the revolution speed of the blower. If the transmission from crankshaft to blower is raised 1.3 fold, i.e., 2.85, the power absorbed by the blower at 1,695 r.p.m. increases to 2.2 times its former value, which is 33.2 hp. or 6 percent of

the sustained engine output of 560 hp. The energy absorbed by the blower is still somewhat smaller than the loss through the resistance of conventional water radiators or of radial engines.

Since the quantity of air delivered increases to 5.79 kg/s (12.76 lb./sec.), the temperature increase of the cooling air drops 11 to 35° C. (19.8 to 63° F.). Since all hot points of the engine are in the vicinity of the outflow of the cooling air, a temperature drop of about 10° C can be assumed. Moreover, the heat transfer is improved by the increased velocity of flow. The rate of heat transfer depends on the heat load of the cooling fins and on the velocity of flow. According to the first section of this report, we can calculate on 1.22 fold coefficient of heat transfer at 1.3 fold velocity of flow for the aluminum fins, on 1.16 fold for the highly loaded steel fins of the cylinder in the vicinity of the combustion chamber and on 1.2 fold in the vicinity of the cylinder base. The mean improvement in the coefficient of heat transfer may therefore be estimated as 1.19 fold. At the temperature of 300° C. (572° F.) in the vicinity of the cooling-air exit, where the temperature difference between cooling-air and the cylinder is therefore 254° C. (457.2°F.) the temperature difference is lowered to about 214° C. (385.2° F.) by the improved coefficient of heat transfer, so that the temperature is about 260° C. (500° F.). From the thermal viewpoint, adequate reliability of operation is thus insured. Even better results are temporarily possible.

The revolution speed of the blower was increased 27 percent by changing the gear wheels. The temperatures of the engine were measured in various test runs. The engine made several short runs with 560 hp. at 1,750 r.p.m. and with 620 hp. at 1,808 r.p.m. For a brief period 678 hp. was attained at 1,820 r.p.m. It was found that the temperatures fell, on the average, about 50° as compared with the runs at the lower revolution speed of the blower. It was also found that the temperatures were lower for 620 hp. at the higher revolution speed of the blower, than for 560 hp. at the lower revolution speed. The observed temperatures are given in the table.

With the present arrangement the exhaust valves are situated behind the intake valves in the direction of flow. The exhaust valves are therefore cooled by the al-

ready heated air. Exchanging these valves would make the exhaust side of the cylinder head 20 to 30° cooler. Increasing the intake side by about the same amount would make the temperature differences in the cylinder head smaller, thus lessening the tendency to warp.

The experiments with the engine also showed that it was very difficult to cool adequately the bell-shaped closed air-cooled cylinder head with the screwed-on head. An exposed cylinder, with an aluminum head shrunk on with a coarse thread, yields better cooling. Even the flanging of the cylinder head on the open liner at the height of the upper piston dead center is open to question. The head can be secured to the cylinder by a larger number of strong screws.

In order to be able to mount the six centrifugal blowers in the space between the cylinder banks, the air flow must be sharply deflected several times, especially at the entrance to the blower, with consequent loss of energy. The air could be delivered, without deflection, to the first pair of cylinders by means of an axial blower. The diameter of the blower wheel would be about 360 mm (14.2 in.) and could be easily installed. The energy required for cooling could be easily reduced.

V. SUMMARY

With the aid of a heating device, the heat transfer to cylinders with conical fins of various forms is determined both for shrouded and exposed cylinders. Simultaneously the pressure drop for overcoming the resistance to the motion of the air between the fins of the enclosed cylinder is measured. Thus the relations between the heat transfer and the energy required for cooling are discovered. The investigations show that the heat transfer in a conducted air flow is much greater than in a free current and that further improvement, as compared with free exposure, is possible through narrower spaces between the fins. Experiments with a large air-cooled single-cylinder engine are then described, which enable comparisons of the energy expended in cooling a radial engine by the relative wind and cooling by blowers with suitable shrouding of the cylinders. The energy required for cooling remains within permissible limits and is smaller than that necessary for overcoming the resistance of an air-cooled

radial engine. Improvement in cooling makes it possible to attain a greater engine output per cylinder. Greater reliability of operation can be attained, because the cooling air can be conducted to the points of greater heat loading. With the aid of the results of the first investigation, the heat loading of the individual parts of the cylinder can be determined from the temperature distribution of the single-cylinder engine and from the quantity and velocity of the air used for cooling. These investigations are of special importance, because there has hitherto been no criterion for higher-speed engines. The heat loading of the cylinder head with 212,000 kcal/m^2/h (78,156 B.t.u./sq.ft./hr.) is about eight times as large as that of the liner. The investigations end with measurements on a 12-cylinder V-engine provided with six centrifugal blowers, each cooling two opposite cylinders.

Translation by Dwight M. Miner,
National Advisory Committee
for Aeronautics.

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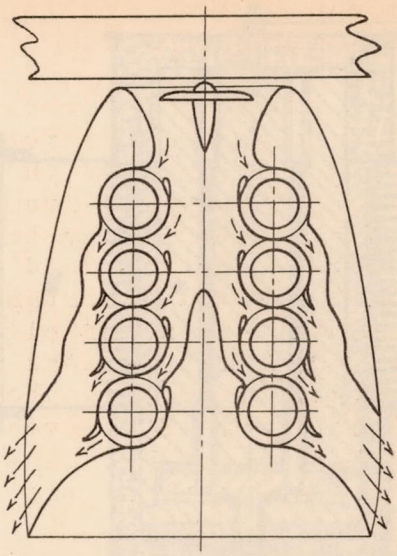


Figure 1.-Eight-cylinder aircraft engine in V form with cooling blower.

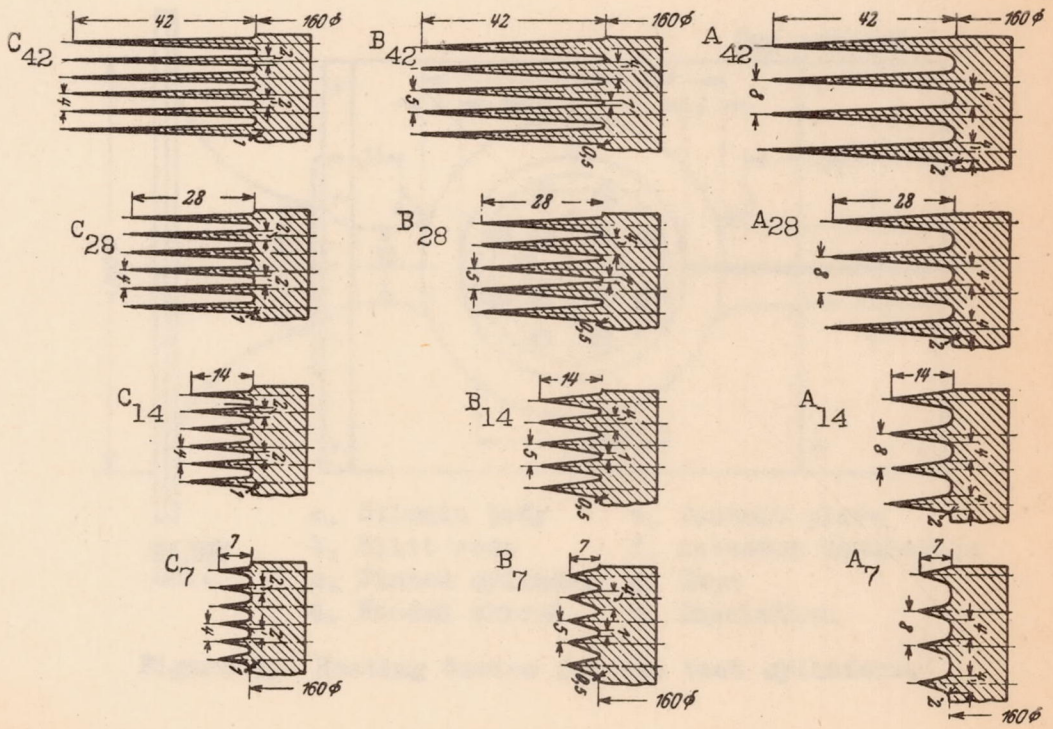
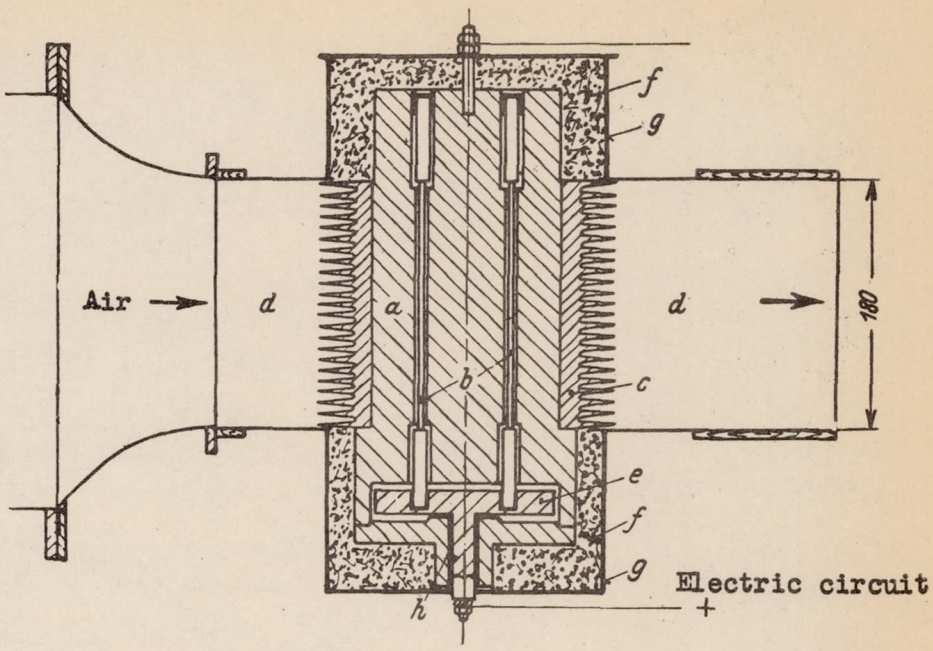
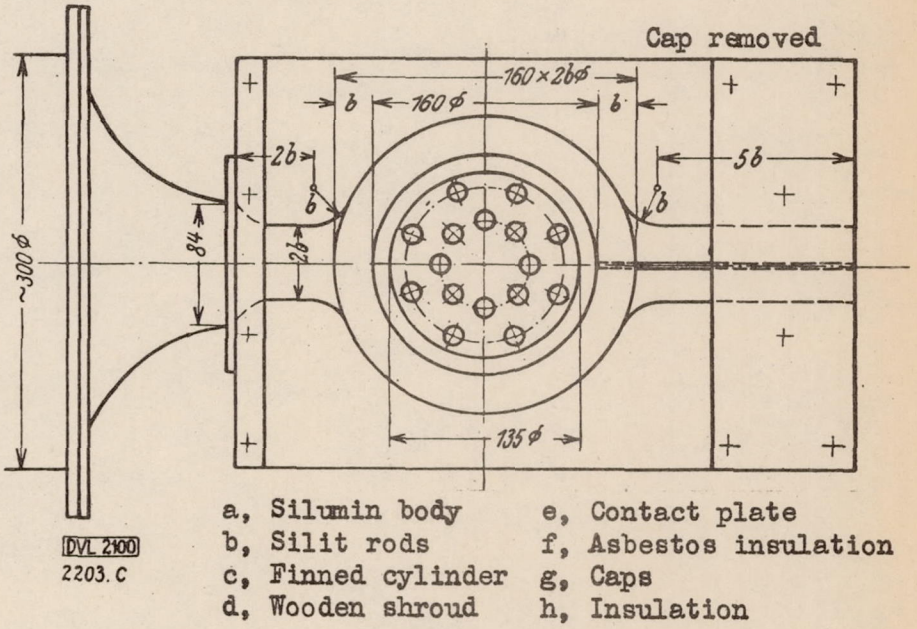


Figure 2.-Fin types used on test cylinders.





a, Silumin body
b, Silit rods
c, Finned cylinder
d, Wooden shroud
e, Contact plate
f, Asbestos insulation
g, Caps
h, Insulation

Figure 3.- Heating device for the test cylinders.

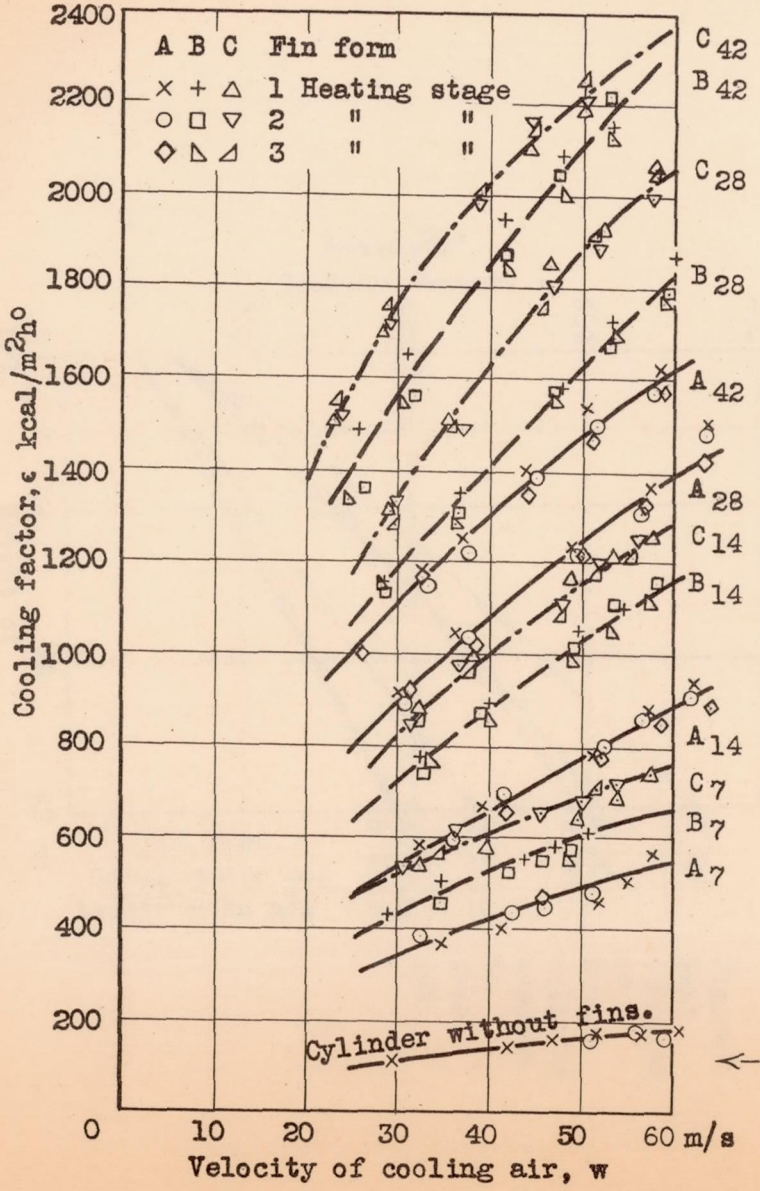


Figure 4.- Heat transfer of shrouded finned cylinder.

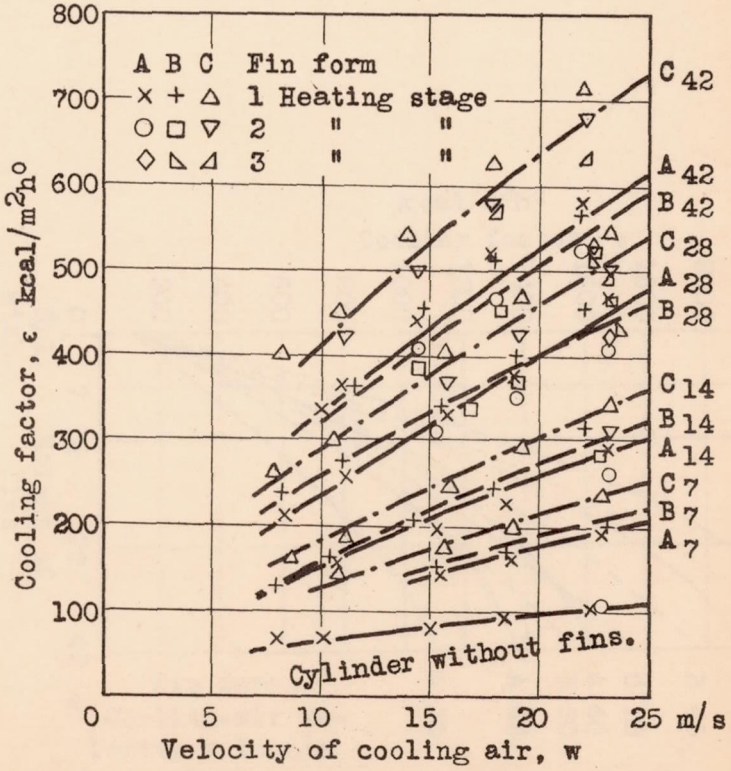


Figure 6.- Heat transfer of finned cylinder exposed to free jet.

Figs. 4,6

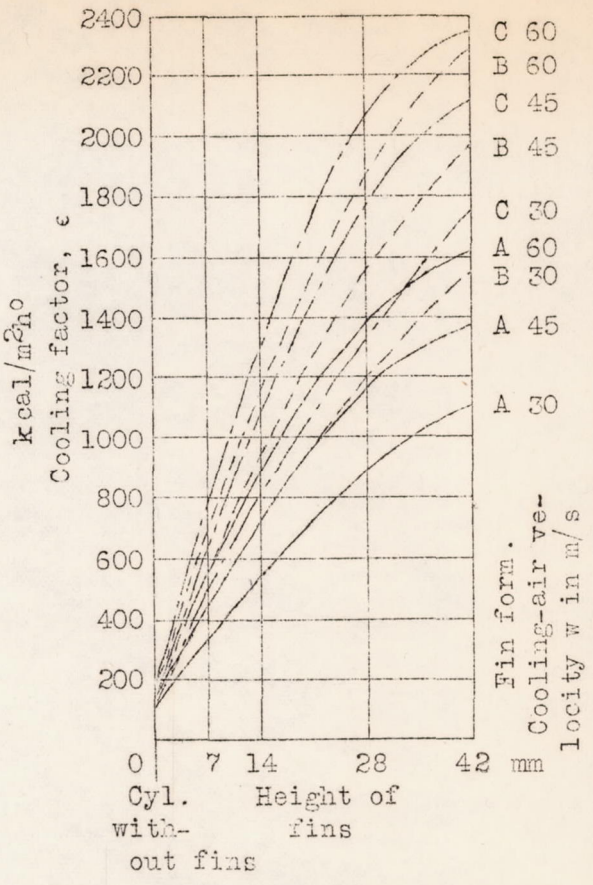


Figure 5.- Heat transfer of shrouded finned cylinder.

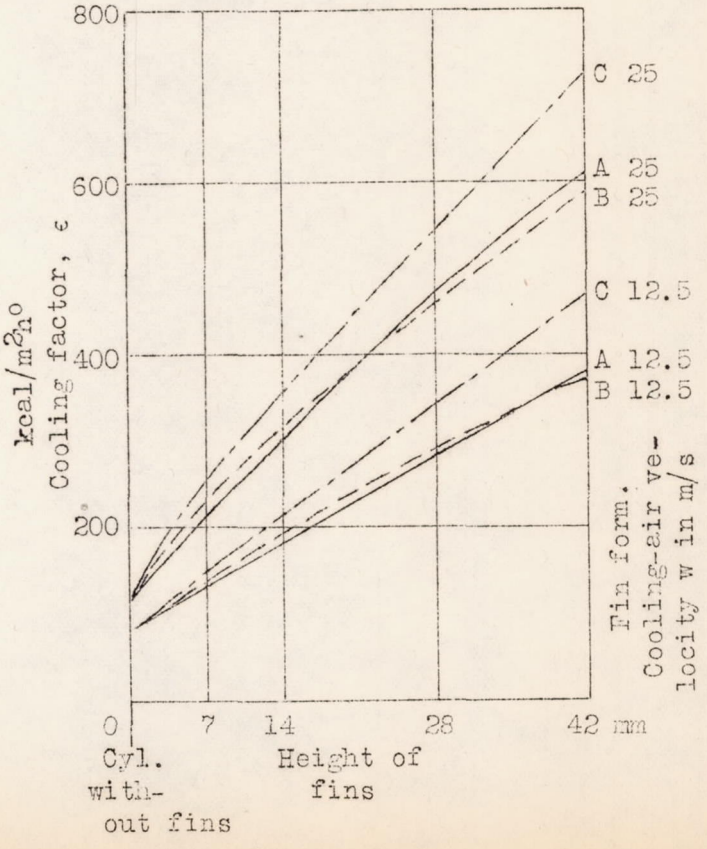


Figure7.- Heat transfer of finned cylinder exposed to free jet.

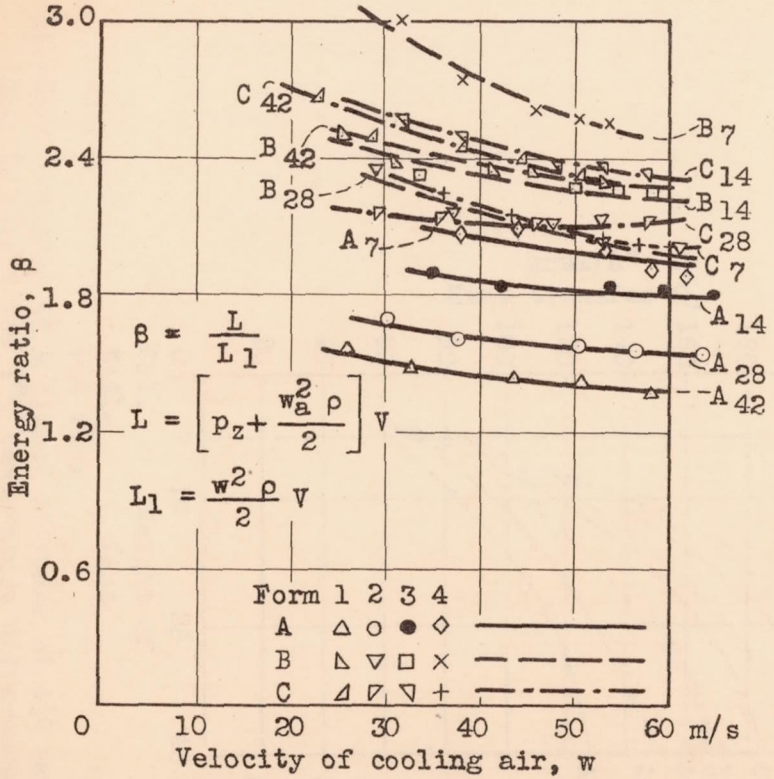


Figure 8.- Energy ratio β plotted against cooling-air velocity w.

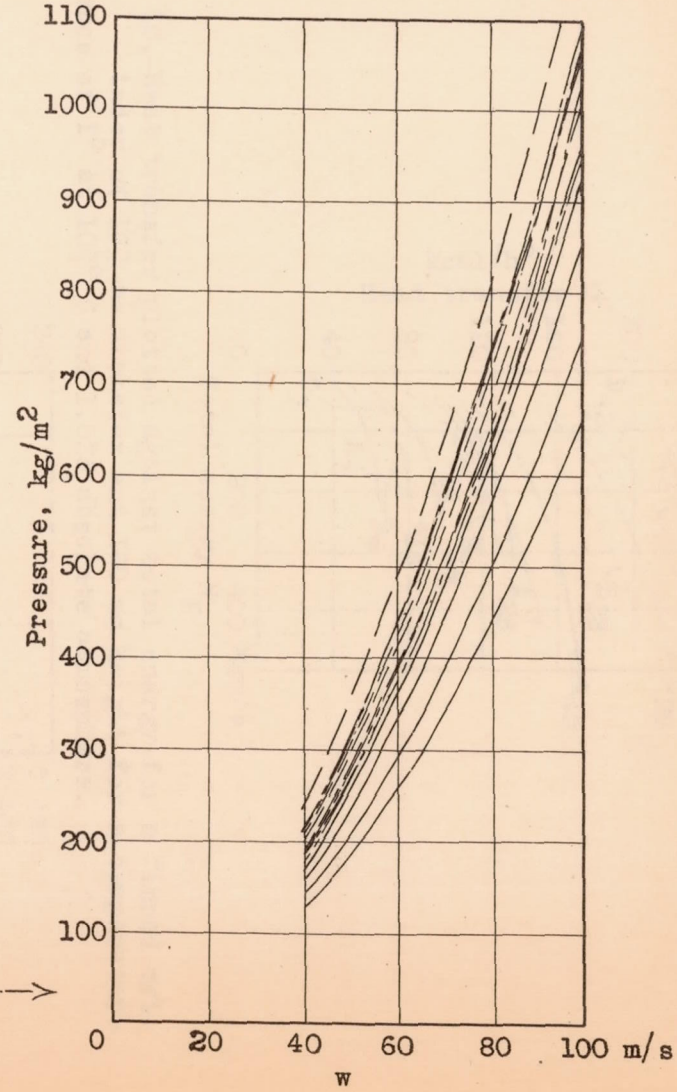


Figure 9.-Total pressure $p_z+w_a^2\,\rho/2$ required to accelerate the air and to overcome the friction and turbulence at 100°C and 1.035 abs.atm. for 160 mm diameter.

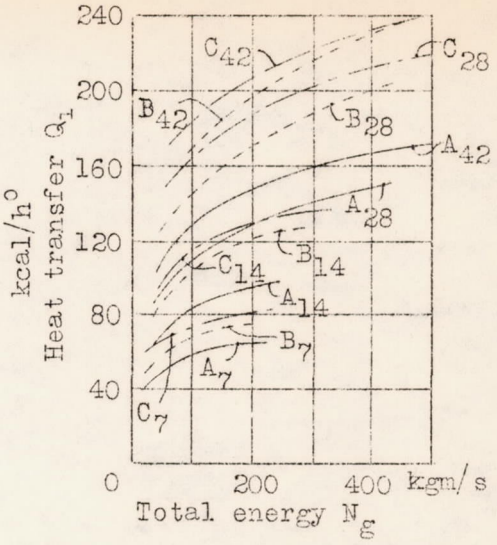


Figure 10.-Heat transfer plotted against total energy for a finned cylinder of 160 mm diameter and 180 mm height for a temperature difference of 1° at 100° C and 1.035 absolute atmosphere.

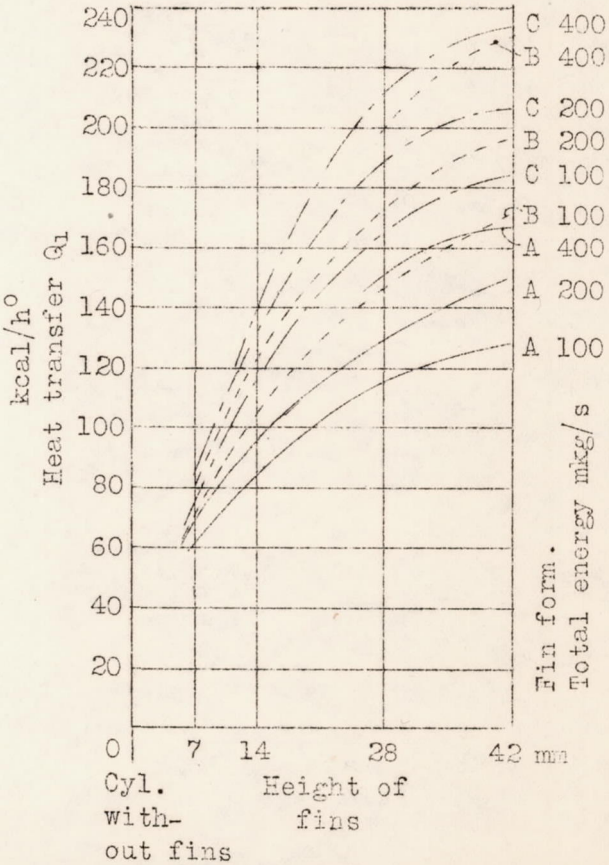


Figure 11.-Heat transfer of a finned cylinder of 160 mm diameter and 180 mm height for 1°C temperature difference at 100°C and 1.035 abs. atm. for 100,200 and 400 mkg/s total energy.

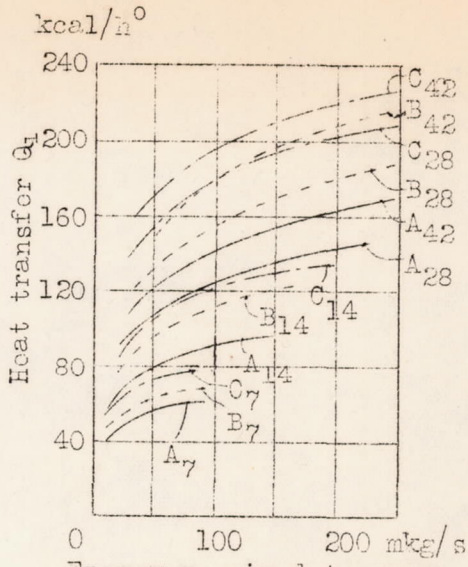


Figure 12.-Heat transfer of finned cylinder plotted against energy required to overcome friction and turbulence, for 160 mm dia. and 180 mm height, for 1°C temperature difference at 100°C and 1.035 absolute atmosphere.

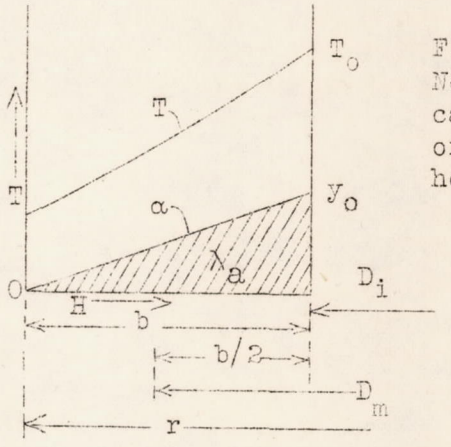
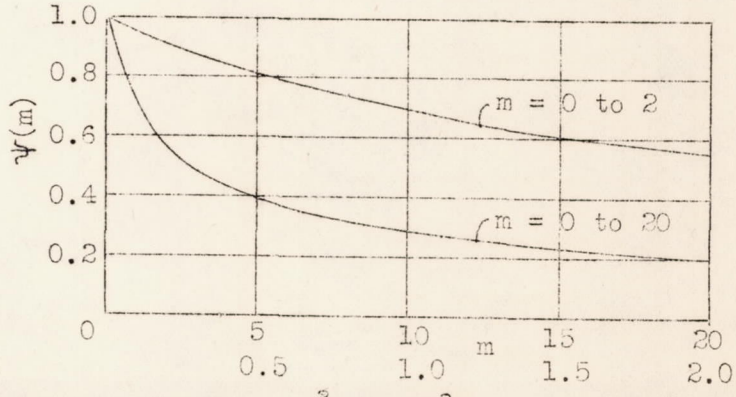


Figure 13.-Notation for calculation of effect of heat on fins.

Figure 14.-Curves for functions $\psi\left(\frac{a_o b^2}{\lambda_o J_o}\right)$ and $\psi_1\left(\frac{a_o b^2 D_m}{\lambda a J_a D_i}\right)$

$\alpha = 22.5\,\lambda/d\,(wd/a)^{0.79}$



$$\psi(m) = \frac{1 + \frac{m}{(1!)^2 2} + \frac{m^2}{(2!)^2 3} + \frac{m^3}{(3!)^2 4} + \cdots + \frac{m^n}{(n!)^2 (n+1)}}{1 + \frac{m}{(1!)^2} + \frac{m^2}{(2!)^2} + \frac{m^3}{(3!)^2} + \cdots + \frac{m^n}{(n!)^2}}$$

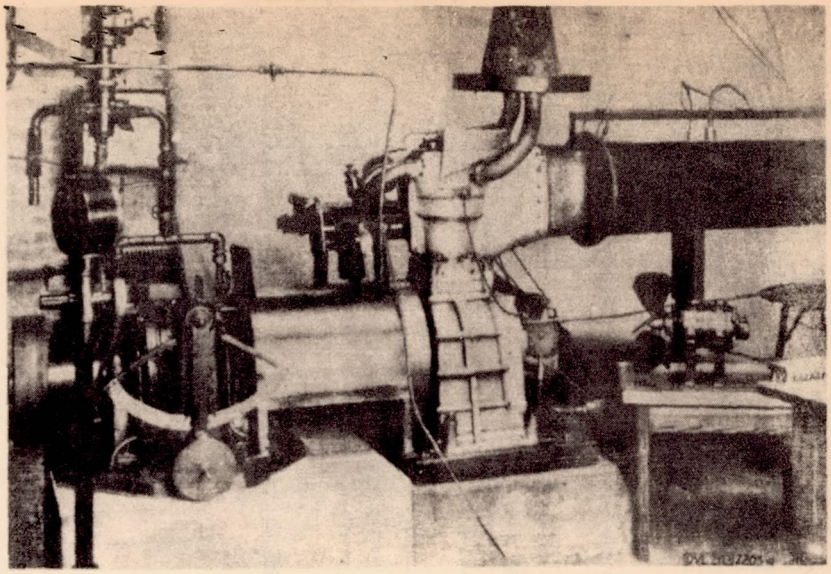


Figure 16.-General view of shrouded single-cylinder engine.

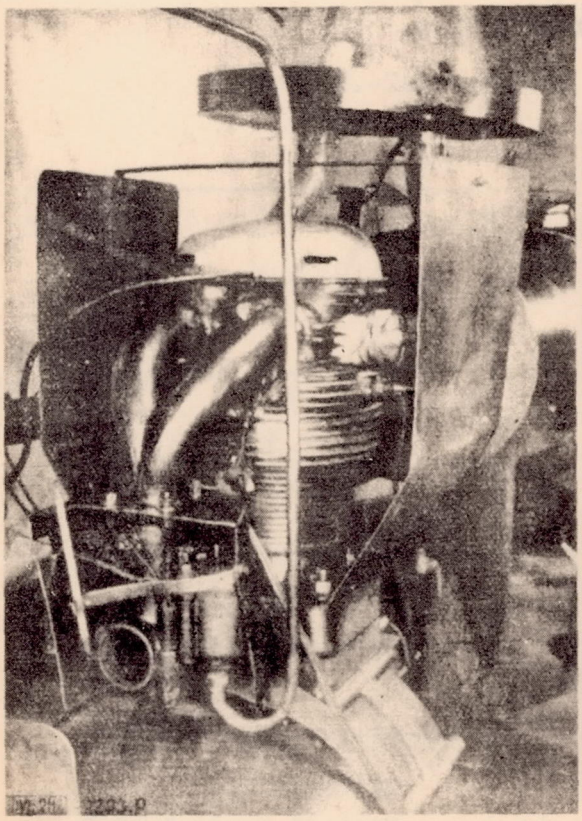


Figure 15.-Single-cylinder engine exposed to free air jet.

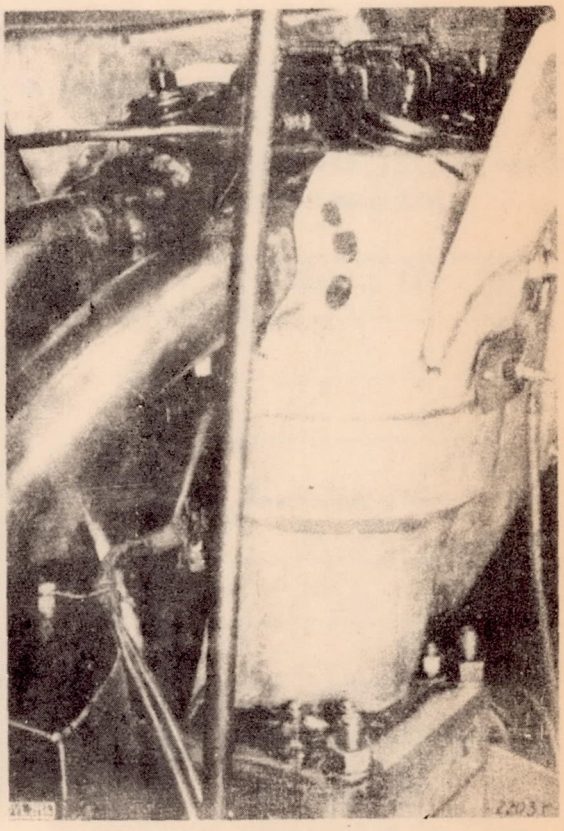


Figure 17.-Shroud of single-cylinder engine.

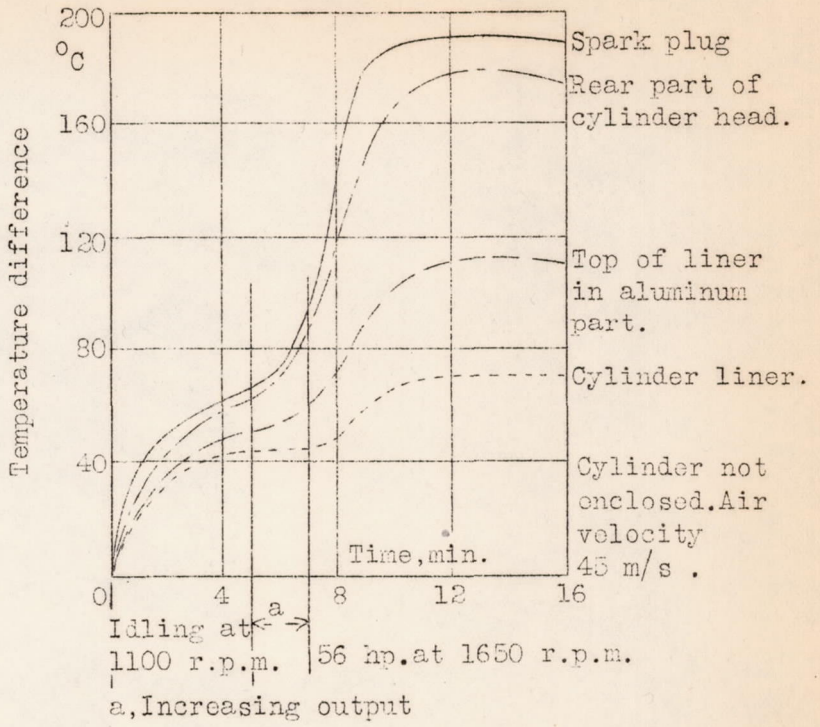
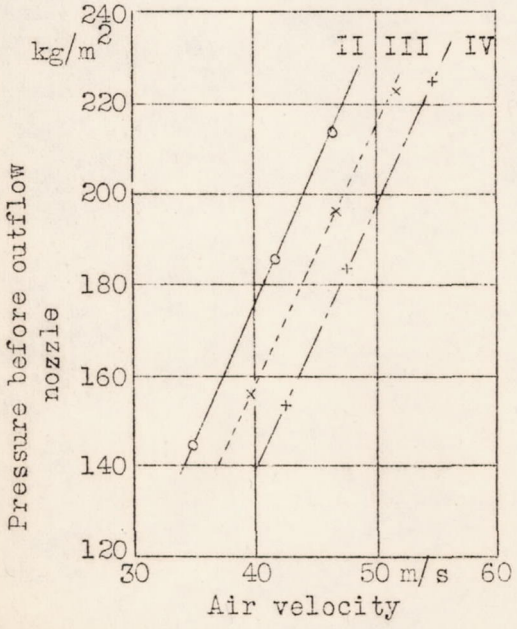


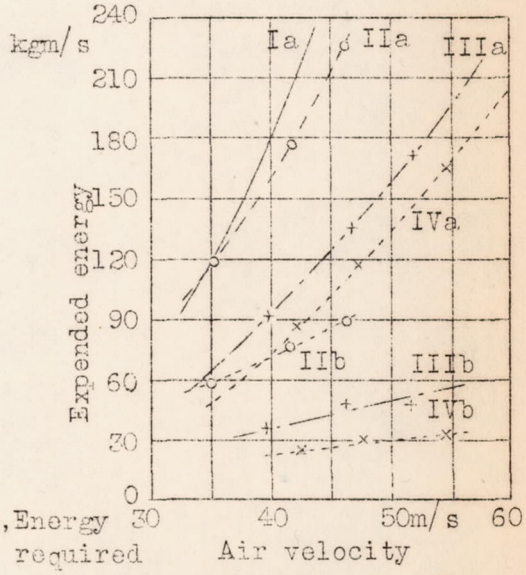
Figure 18.-Temperatures in starting the engine.



II, Shroud 10 mm distant
III, " in contact
IV, Contact shroud with block.

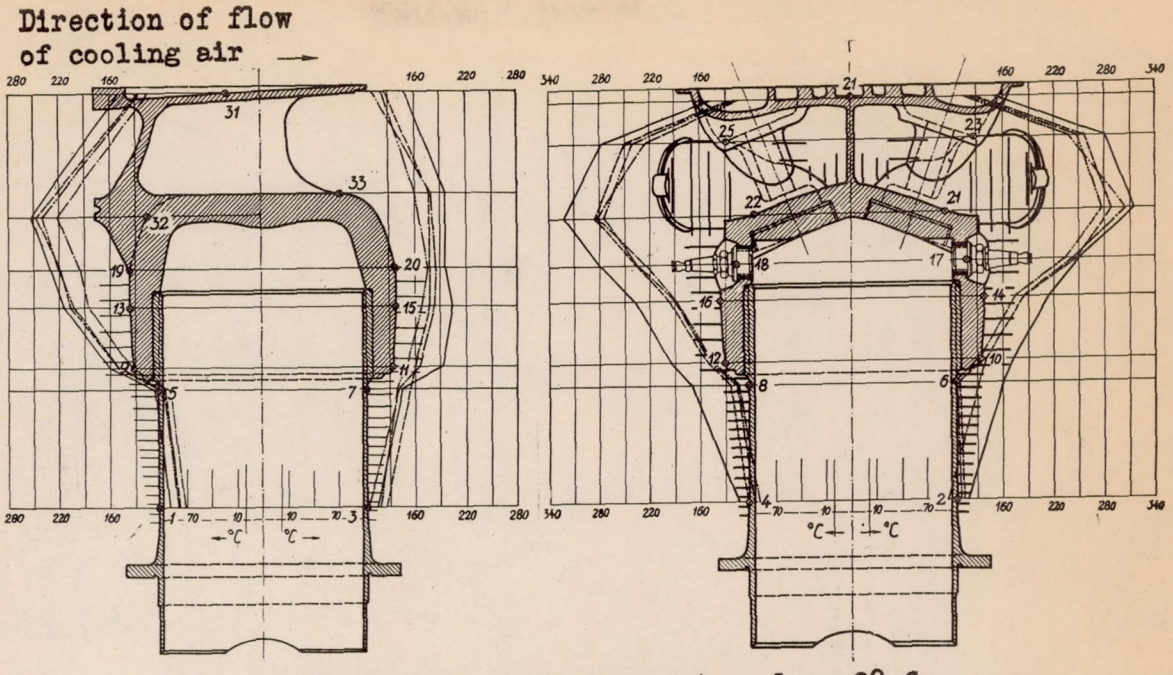
Figure 22.-Velocity and pressure before outflow nozzle in air conduit of single-cylinder engine.

I, Not shrouded. II, III, IV, see Fig. 22
a, Total energy expended.



b, Energy required to overcome resistance to flow.

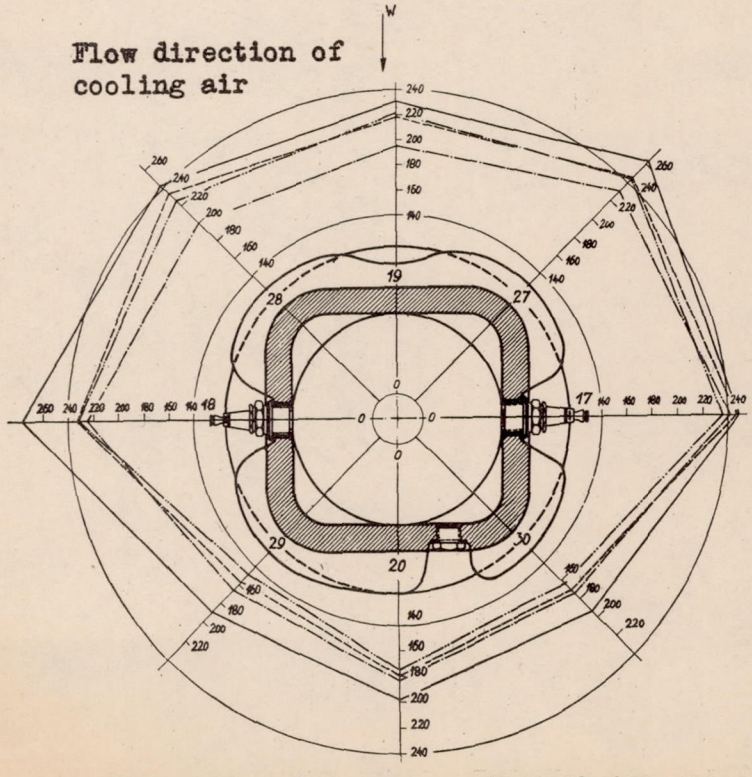
Figure 23.-Expended energy plotted against velocity of air in single-cylinder engine with different shroudings.



Scales, drawing 1:6, temperature 1 mm 9° C
The mark ⊗ indicates exact test point

Figure 19.- Temperature distribution on single-cylinder engine. Section through cylinder axis in direction of flow.

Figure 20.- Temperature distribution on single-cylinder engine. Section through cylinder axis perpendicular to direction of flow.



Not shrouded

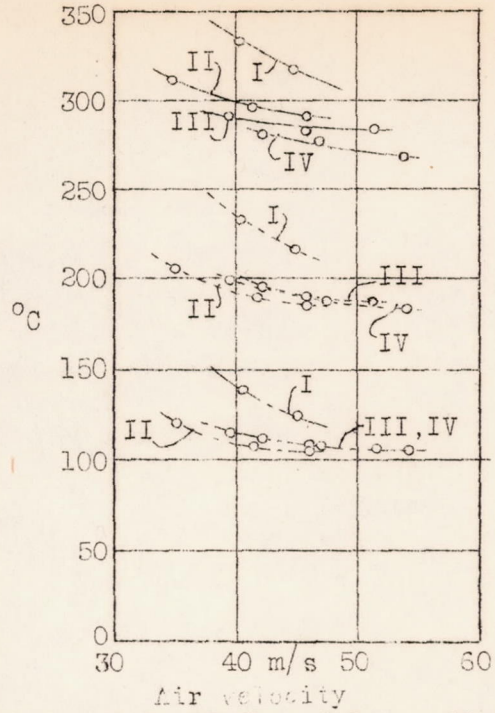
Shroud at 10 mm distance

Shroud in contact

Shroud in contact and wooden block

Scales; drawing 1:6
temperature (plotted from 0) 1 mm 6° C

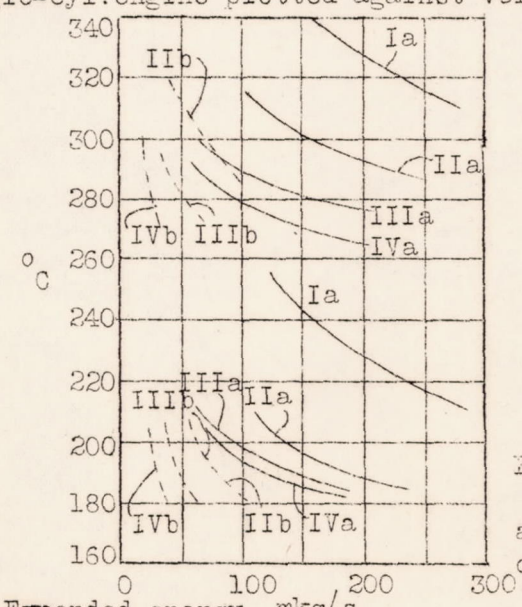
Figure 21.- Temperature distribution on single-cylinder engine. Horizontal section through cylinder head.



——— Maximum temp. between inlet and outlet pipes.
- - - - Mean temp. of cyl. head.
— — — " " " " liner

I,Not shrouded. II,Shroud 10 mm distant. III,Shroud in contact. IV,Contact shroud with block.

Figure 24.-Temperature differences on single-cyl.engine plotted against vel.of air.



I,Not shrouded. II,Shroud 10 mm distant. III,Shroud in contact. IV,Contact shroud with block.

a,Total energy expended. b,Energy required to overcome resistance to flow.

Figure 25.-Temp. difference on single cylinder engine plotted against energy required with different casings.

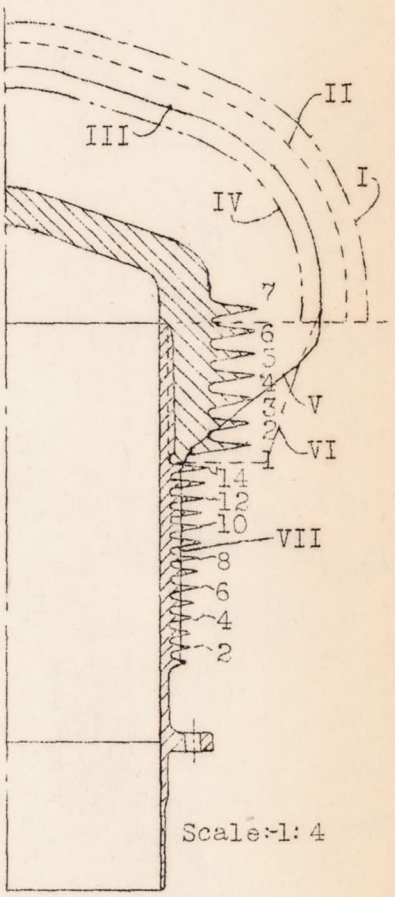


Figure 26.-Heat loading of inner wall of test cylinder.
Scale:-One mm from inner wall corresponds to 10,000 kcal/m^2/h .

——— Max. temp. between inlet and outlet pipes.
- - - - Mean temperature of cylinder head.

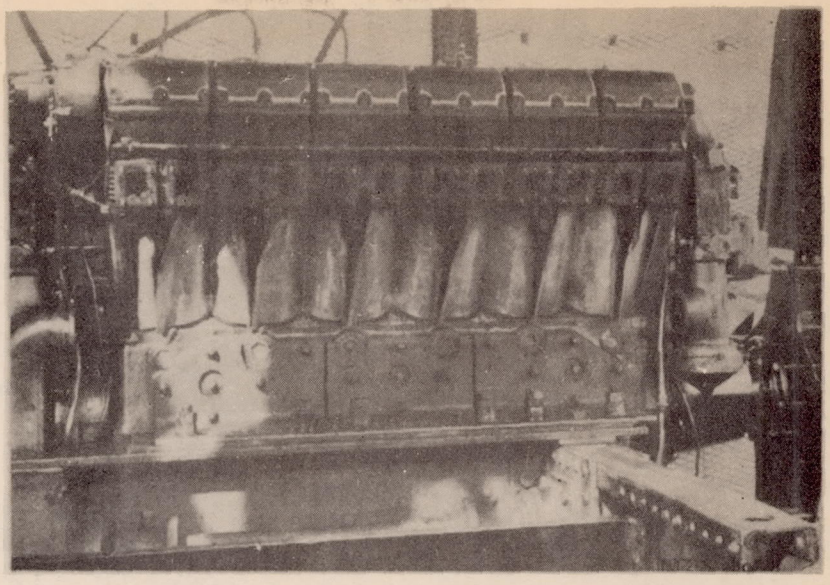


Figure 27.-Outer side of 12 cylinder engine with DVL air guides.

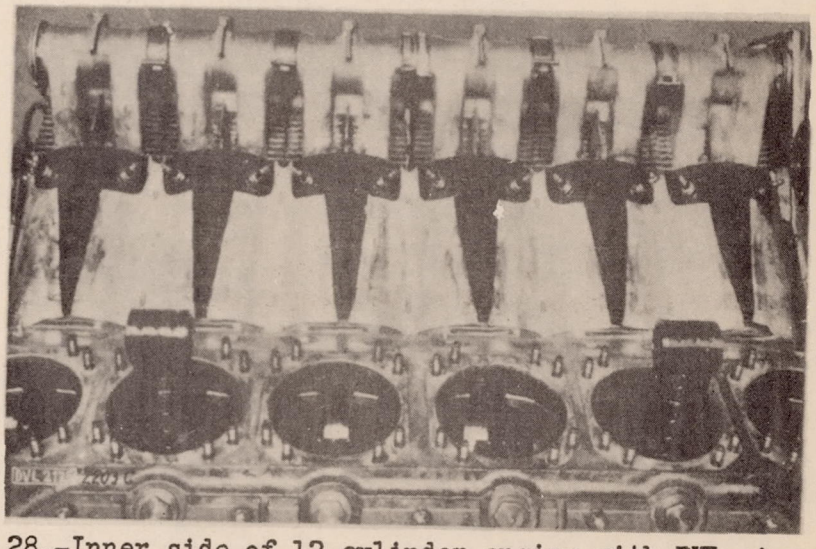


Figure 28.-Inner side of 12 cylinder engine with DVL air guides.
(Blower and left bank of cylinders removed.)

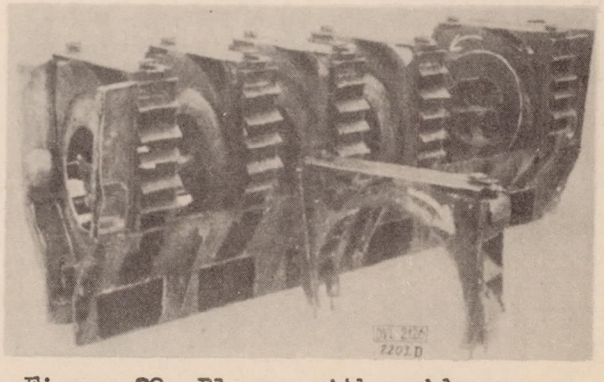
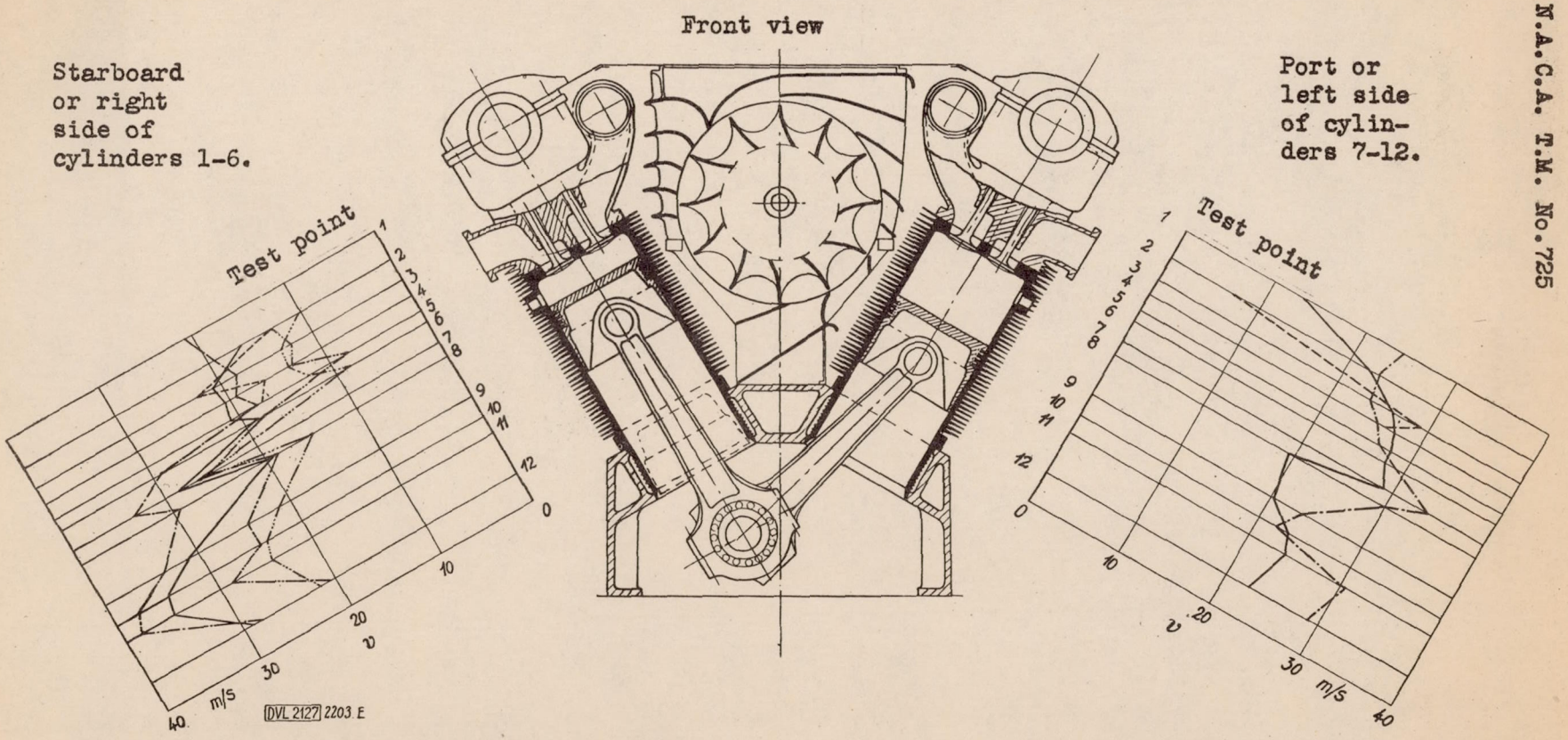


Figure 29.-Blower with guide vanes.



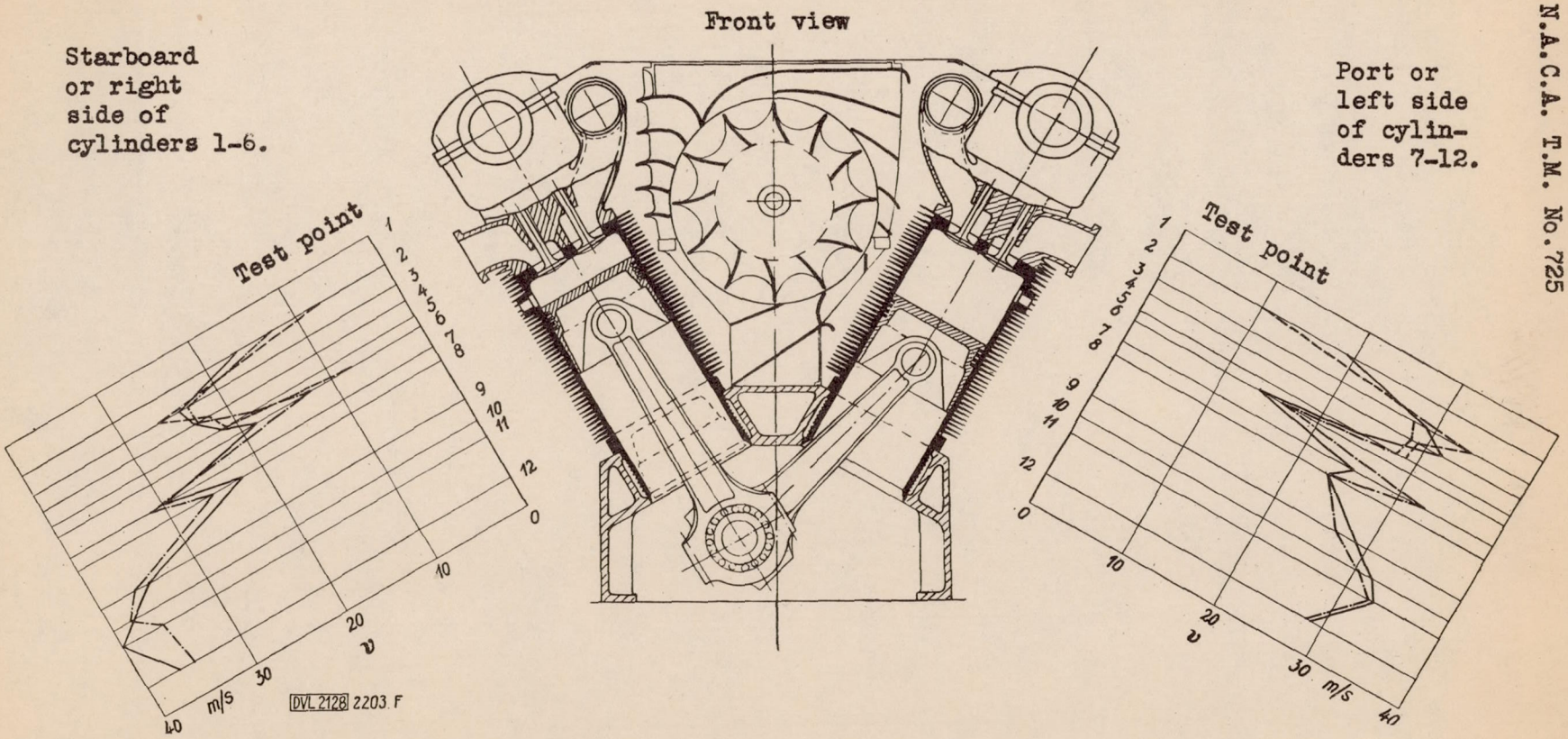
a) With free intake orifice

——————— Between cylinders 2&3 and 8&9
——— - ——— " " 3&4 and 9&10
- - - - - - - " valves on cylinder heads 5 and 1

b) With nozzle over intake orifice

——— - - ——— Between cylinders 2&3
——— - - - - ——— " " 3&4

Figure 30.-Air velocities with cylinders exposed.



———————Between cylinders 2&3 and 8&9
———— – ———— " " 3&4 " 9&10
– – – – – – – " valves on cylinder heads 2&8

Figure 31.-Air velocities for cylinders with DVL shrouds.